Vol. 16, No. 12 December, 1960

W. H. H. .....
L. N. H. .....
C. E. H. .....
W. C. B. .....
J. .....
P. .....
M. B. .....

LUBRICATION ENGINEERING

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

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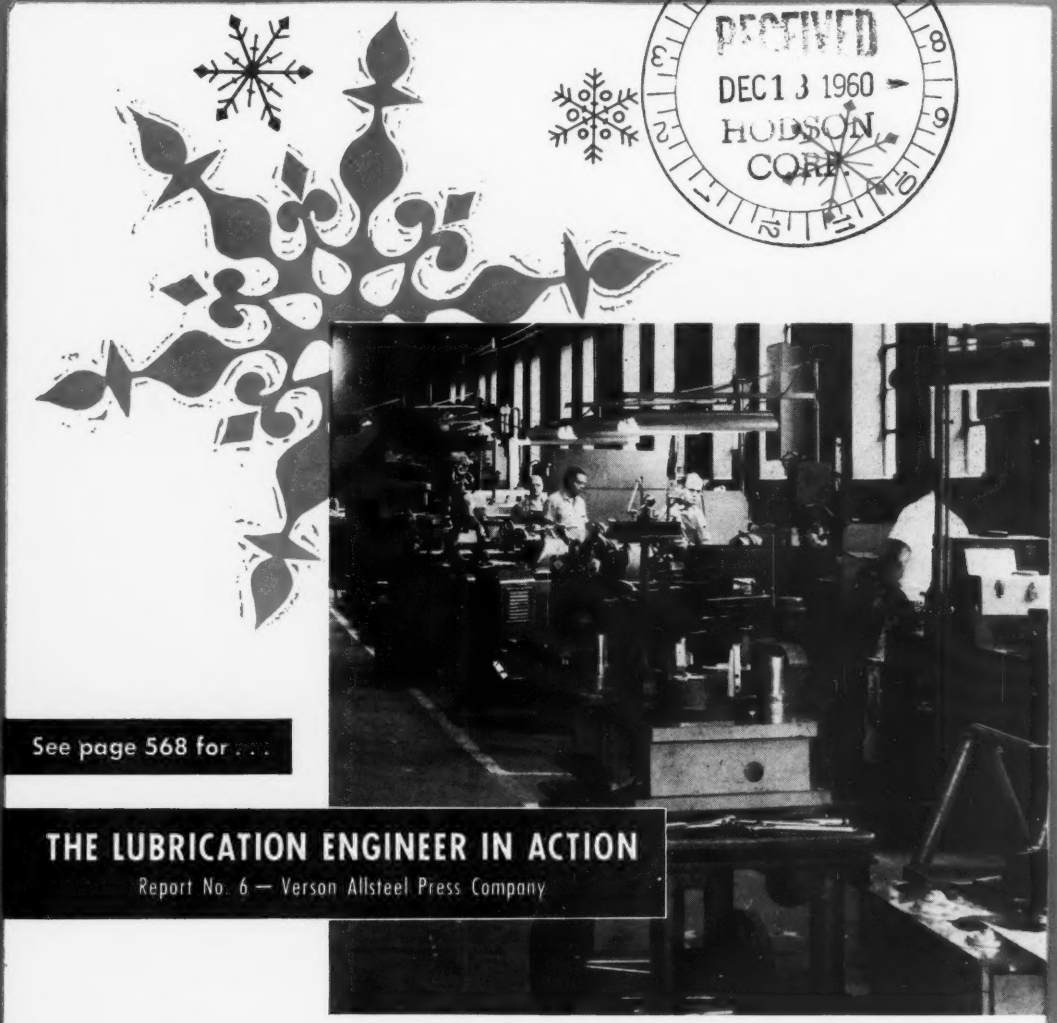

See page 568 for . . .

THE LUBRICATION ENGINEER IN ACTION

Report No. 6 — Verson Allsteel Press Company

JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

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For 84 years, The Brooks Oil Company has supplied superior lubricants to industry throughout the United States and Canada.

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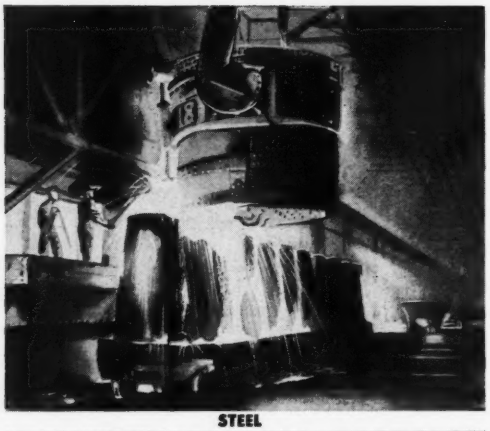
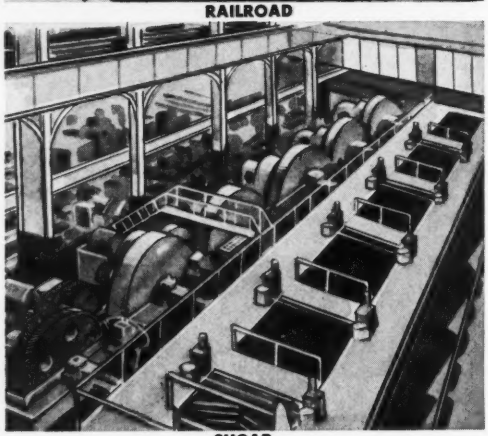
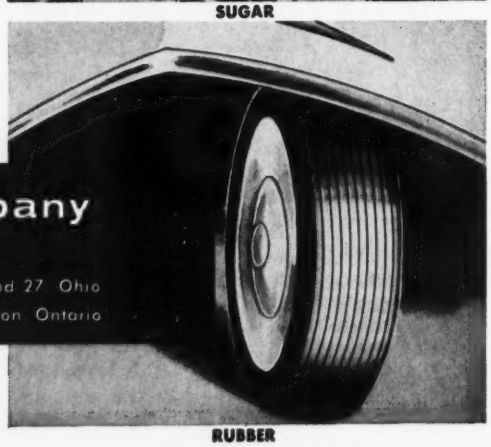


STEEL

RAILROAD



SUGAR



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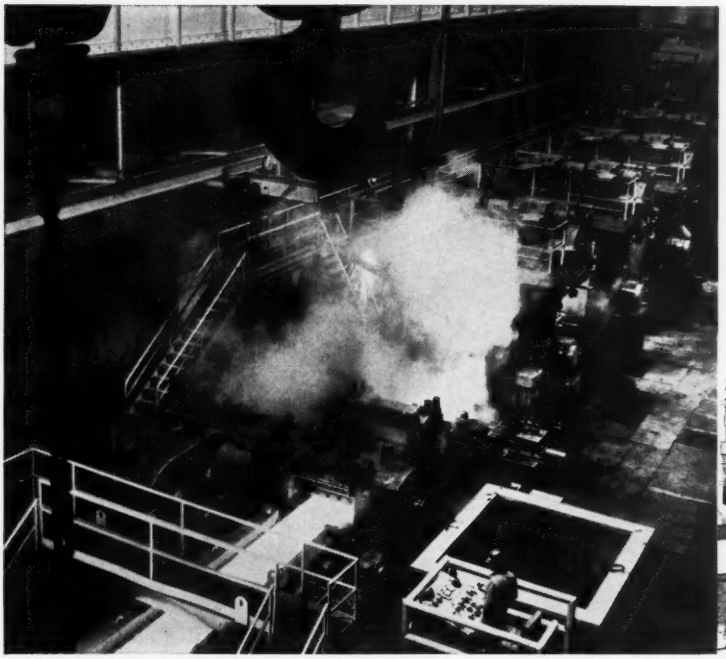
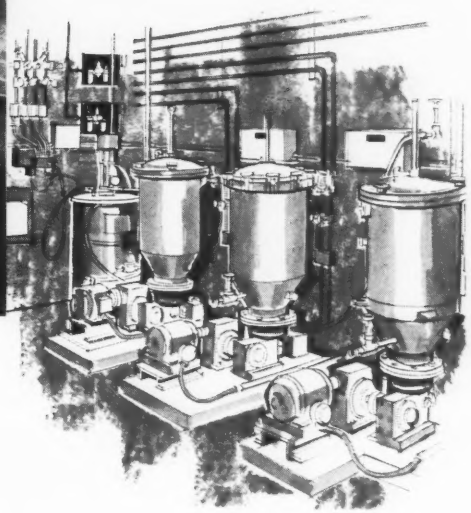


• Typical large, automatic motor-driven pumping station set-up for heavy-duty service on a hot strip mill centralized lubrication system.



Positive Farval centralized lubrication protects 1597 bearings on Jones & Laughlin's hot strip mill

On this semi-automatic, punch card controlled, high-production (*rolls over 100,000 tons monthly*) 44-inch Hot Strip Mill—first of its type in the world—26 Farval centralized lubricating systems work 'round-the-clock to protect 1597 vital bearings—help keep this up-to-the-minute rolling facility in continuous 24-hour production, cut operating costs.

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- Saves time—lubricates all bearings in a few minutes, while mill equipment remains in full operation.
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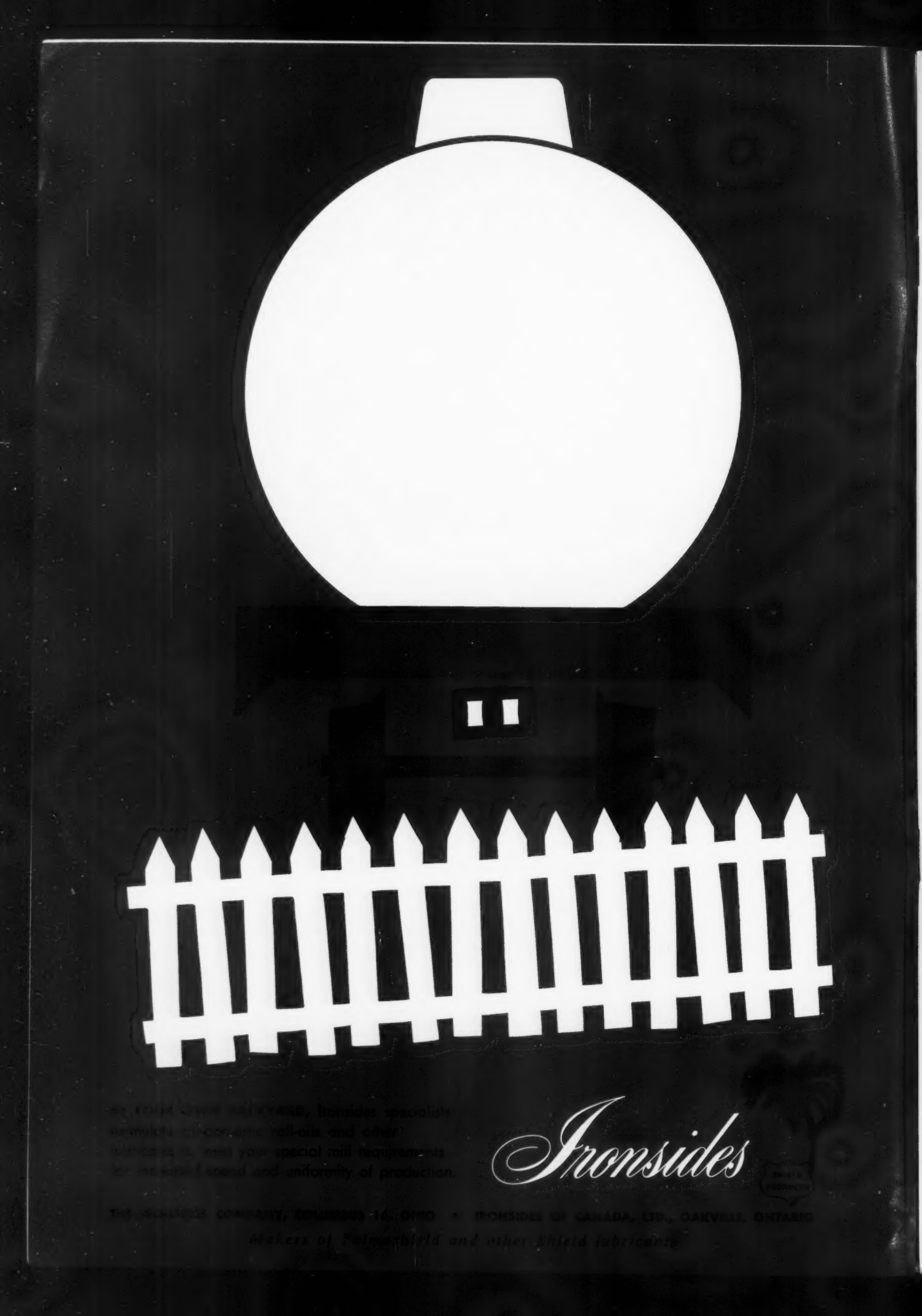
- Saves bearings—bearing life increases 5 to 25 times when Farval serves the equipment.
- Saves lubricant—much as 3 out of every 4 pounds.

Farval can be relied upon to provide the right amount of lubrication—when it's needed, where it's needed. In addition Farval provides indication and adjustment at every bearing, system time cycle flexibility, and handles the widest range of lubricants. Many bearings can easily be lubricated over a large area from one conveniently located central station.

Remember, for the ultimate in bearing protection on all types of industrial production machines and equipment—Farval is the cost-cutting answer. Get the latest information on how Farval can fit into your production picture—it's in free Bulletin 26-T. Write us for your personal copy, today.

Farval Studies in Centralized Lubrication No. 252

Farval Division • Eaton Manufacturing Company
3267 East 80th Street • Cleveland 4, Ohio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Volume 16, No. 12
December, 1960

Lubrication Engineering

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

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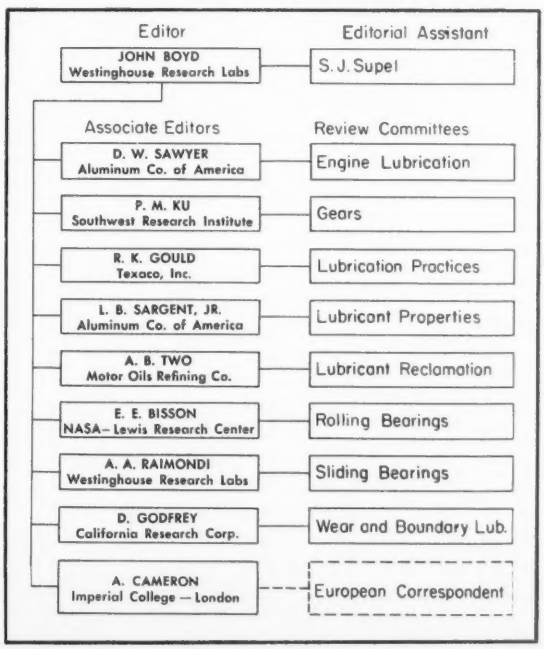

Five AMOCO Additive Product Bulletins to help you

Here are five informative bulletins that give technical data helpful in evaluating AMOCO Additives. Bulletins document with test data and comparative illustrations, the performance of motor oils and gasolines containing AMOCO Additives. Typical physical and chemical properties of the additives are included. Technical service by experienced petroleum additive engineers, to assist you in the use of AMOCO Additives, is also available.

AMOCO 193 INHIBITOR
Gives information on this multi-functional zinc dithiophosphate motor oil additive which inhibits oil oxidation, protects bearings from corrosion, and prevents excessive wear of critical engine parts.

AMOCO 200 SERIES DETERGENT-INHIBITOR ADDITIVES
Data are given on the use of a number of the outstanding detergent-inhibitor additives of this series in making motor oils. All are combinations of Amoco's unique barium detergent and zinc dialkyl dithiophosphate inhibitor.

AMOCO 600 VISCOSITY INDEX IMPROVER
Contains data on how AMOCO 600 VI Improver helps provide easier starting, increased gasoline mileage, less ORI, greater shear stability, better oil mileage. Useful blending chart included.

AMOCO 520 SERIES METAL DEACTIVATORS
Information is provided on extending storage life of copper-contaminated gasoline stocks, heating oils and diesel fuel with metal deactivators of this series.

AMOCO 530 SERIES GASOLINE ANTIOXIDANTS
Each of the three major chemical types of gasoline antioxidants, for economical protection from gum formation, are included in the AMOCO 530 Series. Here are data on all of them.

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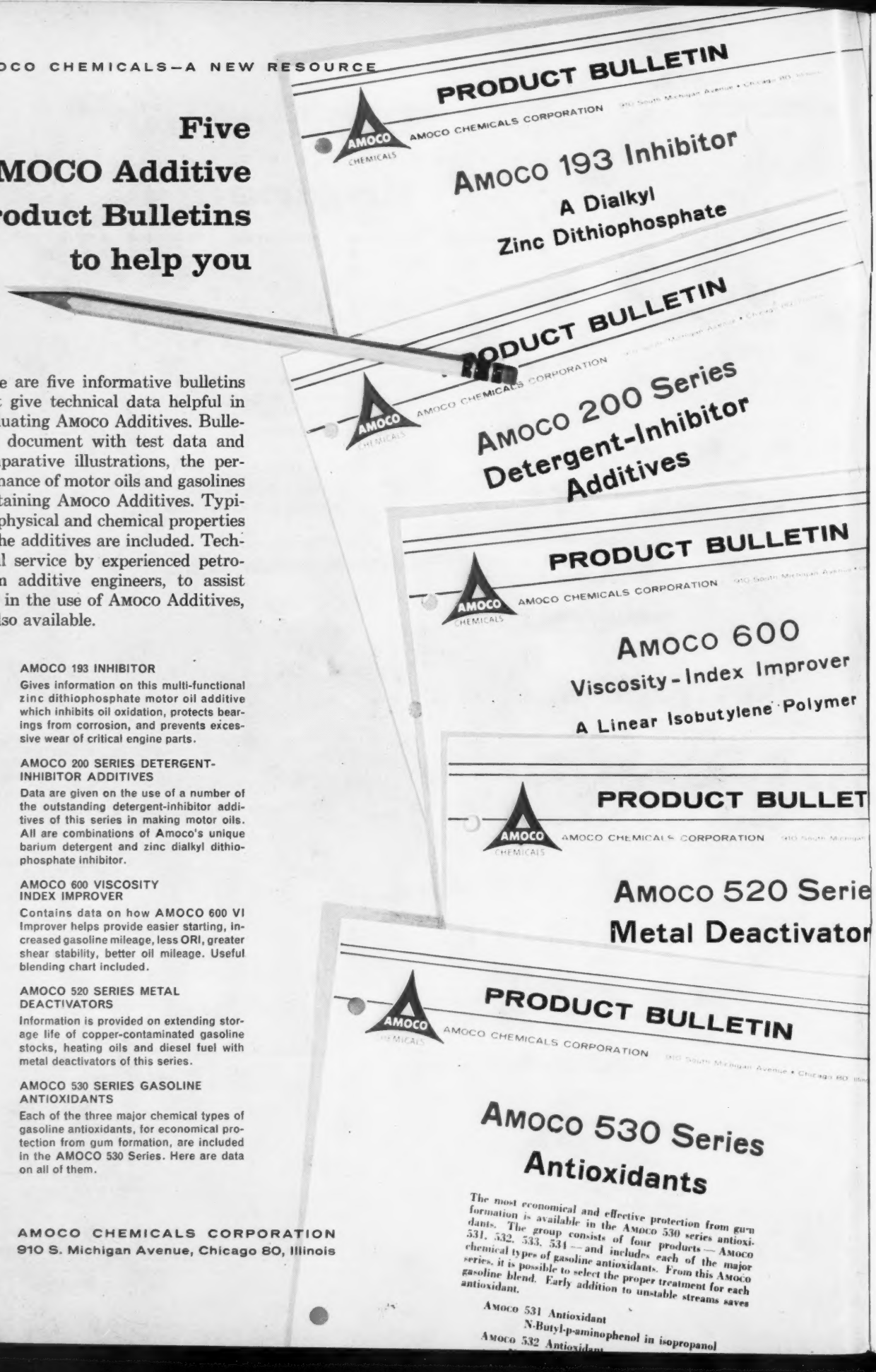
PRODUCT BULLETIN
AMOCO CHEMICALS CORPORATION
AMOCO 193 Inhibitor
A Dialkyl
Zinc Dithiophosphate
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AMOCO 200 Series
Detergent-Inhibitor
Additives
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AMOCO 530 Series
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The most economical and effective protection from gum formation is available in the AMOCO 530 series antioxidants. The group consists of four products — AMOCO 531, 532, 533, 534 — and includes each of the major chemical types of gasoline antioxidants. From this AMOCO series, it is possible to select the proper treatment for each gasoline blend. Early addition to unstable streams saves antioxidant.
AMOCO 531 Antioxidant
N-Butyl-p-aminophenol in isopropanol
AMOCO 532 Antioxidant

Lube Conference Highlights

A record attendance of 424 this year indicated the greatest interest ever in the Joint Lubrication Conference held in Boston, Mass.

The Reception was well attended—Those responsible for the warm occasion are left to right: Mr. R. L. Johnson (Head of Lubrication Section, N.A.S.A., Cleveland, Ohio; Vice Chairman, 1960 Conference; Chairman 1961 Conference), Mr. R. A. Bareiss (Vice President, Lessells and Associates, Inc., Boston, Mass.; Chairman, Conference Local Arrangements Committee), Mr. M. M. Gurgo, Esso Standard Division of Humble Oil Company; Eastern Regional Vice President, ASLE), and Dr. E. G. Jackson (National Research Corp., Boston, Mass.; Chairman, 1960 Conference; National Director, ASLE).

Engineers and scientists from throughout the world attended the conference. Participants represented Germany, England, and The Netherlands. Shown left to right are: front row—Dr. E. G. Jackson, Chairman, 1960 Joint Conference Committee; Dr. W. E. Campbell, Past President, ASLE; Mr. C. L. Willey, Executive Secretary, ASLE; Prof. H. Blok, Discussor, The Technical University, Delft, The Netherlands; Mr. W. F. Sentner, Past Chairman, Boston Section, ASLE; back row—Mr. R. A. Bareiss, Chairman, Conference Local Arrangements Committee; Mr. M. M. Gurgo, Eastern Regional Vice President, ASLE; Dr. G. Lechner, Author, Institute of Technology, Munich, West Germany; Mr. S. R. Calish, Jr., National Director, ASLE.

Dr. A. Nahavondi, Westinghouse Electric Corp., presents his paper, "A Novel Form of Self-Acting Gas Lubricated Thrust Bearing," while A. M. Loeb, Vice Chairman, looks on. (Photo Courtesy: George Kellis, Area Representative, Eastern Region)

Dr. M. E. Merchant, Vice Chairman, Session of Friction and Wear, makes a point. (Photo Courtesy: George Kellis, Area Representative, Eastern Region)

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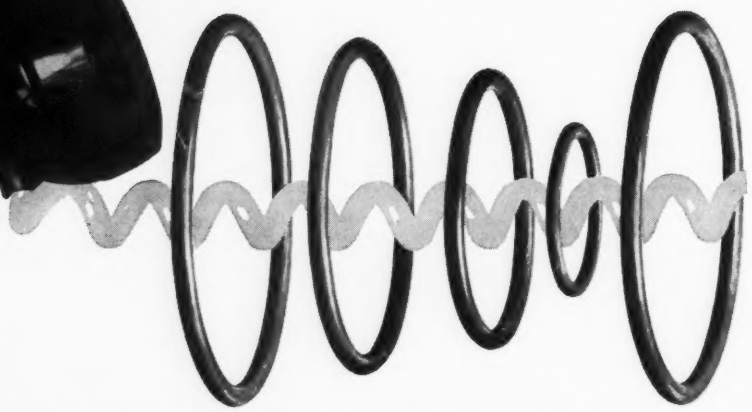

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6342

LUBRICATION IN THE NEWS

The Society will not be responsible for the accuracy of statements made in this column. Any reference to proprietary products does not imply an endorsement of such products by the Society.

HP RATINGS OF AMERICAN STANDARD ROLLER CHAINS PROPOSED FOR ASA ADOPTION

The Association of Roller and Silent Chain Manufacturers has announced the publication of new Horsepower Ratings applicable to American roller chains. The complete horsepower ratings listed in this book are offered as an aid in design by expanding the application field for roller chain drives and open the way for more horsepower per chain dollar.

These ratings represent the results of years of research sponsored by the Association of Roller and Silent Chain Manufacturers in which studies were made of: a) Roller Impact Forces, b) Dynamic Tension Forces, c) Efficiency, d) Wear Life. Each of these studies increased the available knowledge of roller chain characteristics. Wear studies, for example, showed that properly applied lubricant formed a separating film in chain joints in a manner similar to that found in journal bearings. This made possible much more accurate predictions of wear life under known conditions of service. It opened an area of roller chain application at high speeds previously considered impractical. The direct result is increased horsepower ratings—more horsepower capacity per roller chain dollar. (Source: Assoc. of Roller and Silent Chain Mfgs.)

BATTELLE MEMORIAL INSTITUTE CONDUCTS RESEARCH IN BEHAVIOR OF HIGH-SPEED LIQUID LUBRICATED FACE SEALS

A major research program to provide a more basic understanding of factors involved in the design and use of high-speed rotary-shaft face seals has begun at Battelle Memorial Institute. The program is reportedly jointly sponsored by "more than a dozen large American manufacturing companies interested in advancing shaft-seal technology," according to C. M. Allen, specialist in bearings, seals, and lubrication technology at the Columbus, Ohio, research center.

Presently on a one-year schedule, the program will be directed toward the dynamic behavior of high-speed liquid-lubricated face seals. Basic experimental data will be obtained on the performance of such seals, with special emphasis on the behavior of the primary seal elements. By carefully planning and conducting these seal experiments, it will be possible to control and monitor the behavior of the entire seal system and its supporting structure. Analysis of data obtained in these experiments will make it possible to evaluate the importance of the various parameters involved. The program is expected to provide seal-design data of direct value in the design, installation, and application of high-speed rotary-shaft face seals and machinery.

Some of Battelle's more recent studies in the area of various aspects of wear, friction, and lubrication have involved: friction and wear of ceramic and cermet seal materials at high speed and temperature; high-speed shaft seals and materials for helium-cooled nuclear reactor applications; elasto-hydrodynamic lubrication of rolling contact bearings; and high temperature lubrication of rolling and sliding bearings with solid lubricants. (Source: Battelle Memorial Institute)

FRICTION, CREEP, VIBRATION, OTHER METALS PROBLEMS STUDIED

The main effects of high temperatures on sliding metal parts are changes in hardness, surface energy, and tensile strength, according to an Air Force-sponsored study of frictional properties of surfaces conducted with a new high-temperature friction apparatus permitting sliding experiments at temperatures up to 2000°F. A technical report of this recent study has been released to science and industry through the Office of Technical Service, Business and Defense Services Administration, U.S. Department of Commerce.

Friction and Wear at Elevated Temperatures, E. Rabinowicz, Massachusetts Institute of Technology, for Wright Air Development Division, U.S. Air Force. Jan. 1960. 25 pages. (Order PB 161719 from OTS, U.S. Department of Commerce, Washington 25, D.C., 75 cents.) Since the limiting temperatures of effective boundary lubrication of bearing materials appears to be somewhere within the range of 1000-1200F, sliding tests were conducted on bearing materials using a specially constructed high-temperature apparatus to determine the effects of friction at temperatures up to 2000°F. From the experimental results, it appears that the main influence of temperature on friction and wear is through changes of hardness, of surface energy, and of tensile strength. The researchers conclude that by running tests such as they performed in which the same combination is tested in both air and in an inert, oxygen-poor atmosphere, it should be possible to decide the role that oxygen plays in the friction process.

Also available from OTS is a report on shear creep-rupture and other properties of aircraft structural metals and joints at high temperatures: *Investigation of the Compressive, Bearing, and Shear Creep-Rupture Properties of Aircraft Structural Metals and Joints at Elevated Temperatures*, L. A. Yerkovich, Cornell Aeronautical Laboratory, Inc., for Wright Air Development Division, U.S. Air Force, Mar. 1960. 111 pages. (Order PB 161717 from OTS, U.S. Department of Commerce, Washington 25, D.C., $2.50.) Two aircraft structural metals—a vanadium-aluminum-titanium alloy and a steel alloy—were tested in the 700 to 1000°F range to measure their creep and rupture performance under the individual influences of tension, compression, bearing, and shear stresses quite similar to the normal creep behavior of metals subjected to tension. The report also includes a discussion of the creep-rupture characteristics of single-rivet lap joints and two-rivet doubler joints prepared from the test alloys. (Source: OTS, U.S. Department of Commerce)

1961 PLANT MAINTENANCE AND ENGINEERING CONFERENCE TO BE HELD IN CHICAGO JAN. 23-26

The 1961 Plant Maintenance and Engineering Conference will be held during the first three days of the four-day Plant Maintenance and Engineer-

(*Continued on page* 555)

NEW *EMRALON*® TFE COATINGS SOLVE HEAT SENSITIVE APPLICATIONS

Acheson's resin-bonded tetrafluoroethylene (TFE) surface coatings are finding ready acceptance among design engineers. Where previous limitations of pure TFE prevented its application on heat sensitive substrates, because of its 700° F. sintering temperature, 'EMRALON' coatings are being successfully used on rubber, light metals, woods, and plastic. One such impressive case history is outlined here.

'EMRALON' coating is shown being spray-applied on woven belting with conventional equipment.

OVER $15,000.00 ANNUAL SAVINGS PROJECTED BY PROMINENT TIRE MANUFACTURER

Dunlop Tire and Rubber Corporation, Buffalo, N. Y., estimates this considerable savings in down-time costs alone on the basis of nearly nine months production use of an 'EMRALON'-coated belt conveyor. The particular belt involved is thirty feet long. Its importance in this company's tire production is emphasized by the fact that it is located at the point of a previous bottleneck. Rubber strips, conveyed from a series of masticating mills to an extruding press, formerly became stuck to the belting of this short-run conveyor immediately above the final masticator. The reason: the temperature of the rubber is elevated — through friction of the masticating process — to between 150° F. and 300° F. at this point. Belts had to be replaced anywhere from once every several hours to every six weeks. It was not unusual to have two or three stoppages every shift. The result: with the supply of rubber temporarily cut off, the entire department shut down. Costs ran $5.00 a minute, and with these work stoppages averaging 20 minutes each, the accumulated expense became prohibitive. Since April, 1959, the new belt has been in constant operation at this critical point in the masticating line.

To date it has not been removed from the mill . . . ample proof of the extraordinary non-stick, adhesion and flexing qualities of its 'EMRALON' coating!

The woven conveyor belt is a product of Globe Woven Belting Company, Buffalo, N.Y. Application of Acheson's 'EMRALON' to their belting was the outcome of a continuing research program headed by Globe Research Manager, Fred Kohlhagen. It is applied by standard commercial spray equipment to a film thickness of 0.7 mils and then oven-cured at a temperature of 300° F.

According to Mr. Kohlhagen, "This example will be only one of many success stories told about this coating. 'EMRALON'-coated belting can be used anywhere a slick, non-corrosive belting is needed. Those in the baking, canning, food processing, food packaging, and rubber fields should find immediate use for it." We might add that there are many other industries finding it advantageous to use 'EMRALON' . . . yours quite possibly is one of them. Why not write today for further information?

'EMRALON' — coated conveyor belt carrying hot rubber through masticating mill.

'EMRALON' ADVANTAGES PROVED IN MANY LIGHT-LOAD USES

The growing list of production applications for the new Acheson 'EMRALON' surface coatings, is convincing proof of their versatility. That film pack, office machine, and lock mechanisms; rubber molds and oil seals; bakery dough handling equipment and countless other light load, solid-film lubrication problems have successfully been answered, is now a matter of record.

With 'EMRALON' 310* — colloidal TFE in a phenolic resin solution — a one-hour cure at 300° F. is all that is necessary. And, for substrates even more sensitive to heat, 'EMRALON' 320** colloidal TFE in a thermo-plastic resin carrier — is available. Both provide excellent adhesion, low coefficient of friction, flexibility, toughness, and corrosion resistance. Applied by conventional paint spray equipment, they allow easy production line conversion to 'EMRALON' solid-film lubricants.

'EMRALON' coatings represent the latest in a long line of widely accepted Acheson products, one of which may offer a ready-made answer to your particular lubrication problem. We welcome the opportunity to be of service. Send for your copy of our Products List which outlines the full line of Acheson Dispersions. Address Dept. LE-120.

*'EMRALON' 310 is manufactured under exclusive license from E. I. duPont deNemours & Company (Inc.) under U. S. Patent 2,825,706. Not licensed for use or for sale for use in providing electrical insulation.

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December, 1960

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LUBE IN THE NEWS

(*Continued from page* 551)

ing Show in Chicago Jan. 23 through 26. The exposition and morning sessions of the conference will be held at the International Amphitheatre. Evening sessions of the conference will be held at the Palmer House. The theme for the show and conference is "The Maintenance Operation Meets the Needs of Increased Production."

In addition to balancing the managerial and technical aspects of the conference, "problem sessions" for 11 specific industries and a discussion of "Insuring Cleanliness in the 'White Room'" will be included in the program. White rooms are those where extraordinary precautions against contamination must be taken.

Topics for the major sessions are "The Design, Application and Results of Our Plant Maintenance and Engineering System," "Case Reports in Preventive Maintenance," "Managerial Controls for Plant and Maintenance Engineers" and "Evaluation of Plant Engineering and Maintenance Performance." (Source: Banner and Greif)

NLGI ELECTS OFFICERS

F. R. Hart is the 27th president of the National Lubricating Grease Institute. Mr. Hart, the supervisor of industrial and marine lubricants for the Standard Oil Company of California, San Francisco, was elected to head NLGI next year during the annual meeting at the Edgewater Beach Hotel in Chicago, November 1. He had previously served as vice president and program chairman.

Vice president of the Institute is C. L. Johnson, president of Jesco Lubricants Company, Kansas City. He had previously served as secretary.

T. F. Shaffer, manager, industrial products department, Shell Oil Company, New York, was elected secretary. Serving as treasurer for his ninth consecutive term is A. J. Daniel, president of Battenfeld Grease and Oil Corporation of Kansas City. L. C. Brunstrum of Standard Oil Company (Indiana), Whiting, Indiana, was reappointed to his second term as chairman of NLGI's technical committee.

Seven directors re-elected to the board for three-year terms include Daniel and Shaffer, as well as S. C. M. Ambler, British American Oil Company, Ltd., Toronto; G. A. Olsen, Sunland Refining Corporation, Fresno, California; F. E. Rosenstiehl, Texaco, Inc., New York; F. W. Minor, Sinclair Refining Company, New York; and W. H. Saunders, Jr., International Lubricant Corporation, New Orleans.

Elected for shorter terms to fill vacancies on the 21-man governing body were N. D. Williams, Pure Oil Company, Chicago; and H. B. Elliott, Atlantic Refining Company, Philadelphia.

In addition to technical marketing programs, NLGI conducts research programs and production surveys for its international membership. T. W. H. Miller is the general manager. (Source: National Lubricating Grease Institute)

INDUSTRY NOTES

Mr. F. L. Townsend, Sales Manager of Wm. W. Nugent & Co., Inc., Skokie, Illinois, announces the appointment of T. F. Hudgins and Assoc., Inc., 3202 Mercer Street, Houston, Texas, as their exclusive representative in the States of Louisiana, Alabama and Mississippi.

The Alpha-Molykote Corp., Stamford, Connecticut, has announced the establishment of a branch office and technical service center at 2645 West Peterson Ave., Chicago. Alpha-Molykote Corp. produces molybdenum disulfide specialty lubricants. Mr. R. B. Dost has been named Midwest District Manager.

It has been announced that Foote Brothers Gear and Machine Corporation has acquired the gear reducer business of David Brown Corporation, Ltd., London. Foote Brothers has been designated exclusive distributors in the United States for speed reducers and gear products manufactured by David Brown Industries of England.

A consolidation and centralization of all ball manufacturing, engineering, sales, and administrative facilities of the Hoover Ball and Bearing Company have consolidated and will be known as the Hoover Ball Division with headquarters at Middletown, Ohio.

from COSMETICS to COMPRESSORS you get top performance with...

UCON
BRAND
FLUIDS AND LUBRICANTS

Whether your product or process calls for a water-soluble cosmetic ingredient or a sulfur-free cylinder lubricant for hydrocarbon gas compressors, UCON fluids and lubricants do the job.

UCON fluids and lubricants also give top performance in many other applications because of these outstanding properties:

- high viscosity indexes
- low pour points
- excellent lubricity
- non-corrosive to metals
- sludge and gum resistant
- sulfur-free
- safe to handle

UCON fluids and lubricants are available in both water-soluble and water-insoluble series with wide viscosity ranges.

Find out how these outstanding properties can improve your product or process. Write for F-6500.

Union Carbide Chemicals Company, Division of Union Carbide Corporation, 270 Park Avenue, New York 17, N.Y.

UCON is a registered trade mark.

UNION CARBIDE CHEMICALS COMPANY

UNION CARBIDE

Better Greases Through BAROID RESEARCH

BARAGEL 24* provides greater stability in greases made from non-petroleum oils

BARAGEL 24, a non-soap grease-gelling agent, gives greater stability to greases made from such non-petroleum oils as polyalkylene glycols, di-2-ethylhexyl sebacate, di-2-ethylhexyl adipate and castor oil. The development of BARAGEL 24 is another result of Baroid Chemicals' research.

Greases compounded with BARAGEL 24 are subjected to the most rigid testing. One of these tests measures grease penetration over the range of unworked to 10,000 strokes. In non-petroleum based oils, BARAGEL 24 produces greases that are exceptionally stable to working in the ASTM grease worker. These greases show minimum breakdown under the most rigorous operating conditions.

BARAGEL 24 is a reaction product of organic ammonium halides and refined sodium montmorillonite. BARAGEL 24 is a member of the BARAGEL-BENTONE* family of products which has been used so successfully in greases, plastics, paints, inks and other fields.

*Trademark of National Lead Company for an organic ammonium montmorillonite

6074

BAROID CHEMICALS, INC.

A SUBSIDIARY OF
NATIONAL LEAD COMPANY

1809 SOUTH COAST BUILDING
HOUSTON 2, TEXAS

NEW PRODUCTS

NEW VACUUM BEARINGS AND SOLID LUBRICANTS MARKETED

CBS Laboratories is marketing a new type of vacuum bearing and dry film lubricant for the aircraft and missile industries.

Vacuum bearings processed by CBS Laboratories reportedly perform with very low friction, in high vacuum approaching that of outer space, for periods of time in excess of 1,500 hours.

CBS Laboratories Dry Film Lubricants, as applied to ball bearings or other frictional surfaces, is developed for application in both vacuum and non-vacuum environments. The manufacturer points out the following as some special areas of use: space vehicle components, rotating or sliding surfaces in vacuum tubes, innertial guidance gyro bearings, sealed (non-replenishable) bearings, and lubrication of frictional surfaces, such as gears, cams, and other sliding surfaces where accessibility is restricted.

Circle No. 1 on Reader Service Card

NEW LOW-COST PILLOW BLOCK UNITS DEVELOPED

The Fafnir Bearing Company of New Britain, Connecticut has announced a new low-cost pillow block unit which incorporates high quality bearings. Known as the PBS, this new unit accommodates shaft sizes between ½ and 1-7/16 inches. It is suitable for most pillow block bearing applications where moderate speeds, moderate radial loads, and relatively light thrust loads are encountered, as in fans, idler shafts, conveyors, pump shafts, and other lightly loaded power shafts.

Circle No. 2 on Reader Service Card

NEW BALL VALVE ANNOUNCED

A new ball valve line, Lunkenheimer Fig. 700, engineered for simplicity of design and maintenance-cutting operation, is being introduced by the Lunkenheimer Co. in bronze body, screw ends, pipe sizes one-quarter to two inches, with Teflon or Buna-N seats and seals.

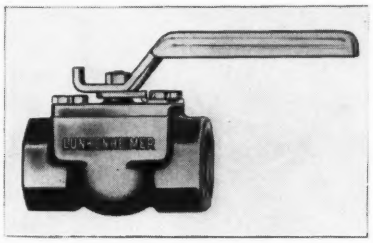


Fig. 700 has top entry through a removable cover, exposing the ball, which is easily removed by the tool on the valve's handle. The ball is flat on both top and bottom, enabling it to be removed from the body for easy access to the seats in case they should need replacement — without breaking the pipe.

The self-aligning ball rides on its seats, sealing at high or low pressure without springs or bearings. O-Ring seals of Teflon or Buna-N protect against leakage at stem and cover.

Circle No. 3 on Reader Service Card

PRECISION PRESSURE REGULATOR MANUFACTURED

A new precision gas regulator is offered by Del Monte Technical Associates to control pressure at any selected remote point in the pumping system. Originally designed for Helium Cryostat service, the Mark II Pressure Regulator is a self-contained, instrument quality device which can be used in both vacuum and pressure applications. Regulation is typically within —0.1 mm of Mercury (0.002 psi.) from minute rates of flow up to 30 cfm. The regulator is reportedly self-correcting for effects of static pressure on the valving, and is further compensated for dynamic forces induced by the gas flow. The Mark II is designed to control pressure at a point upstream. A downstream control version, the Mark IIa, is also available.

Circle No. 4 on Reader Service Card

VARI-TORQUE CONVEYOR AVAILABLE

A new, versatile, chain driven, live roller conveyor, called Vari-Torque Roller Conveyor is now available from Harry J. Ferguson Co. The conveyor has a forward thrust proportional to the weight of the load; transports loads that can be stopped at any point without stopping the conveyor even on a powered curve; permits loads to accumulate and assemble for long periods of time without wear on the rollers and which handles a wide range of items from light to heavy.

Circle No. 5 on Reader Service Card

NEW METAL SEAL FITTING DEVELOPED

Development of a metal sealing ring to use on straight thread fittings is announced by Parker Fitting and Hose Division, Parker-Hannifin Corp. The metal sealing ring is designed to work in the standard SAE O-ring fitting boss, providing a metal-to-metal seating for a highly satisfactory seal.

Made of type 304 stainless steel, the metal ring is offered as an answer to problems in handling high temperatures and rubber-attacking fluids as well as high pressures.

The metal ring can be provided on straight thread steel and stainless steel Triple-lok flared tube fittings, Ferulok "bite" type tube fittings, adapter fittings, and Hoze-lok reusable fittings for industrial hose.

Circle No. 6 on Reader Service Card

NEW SOLID FILM LUBRICANT OFFERED

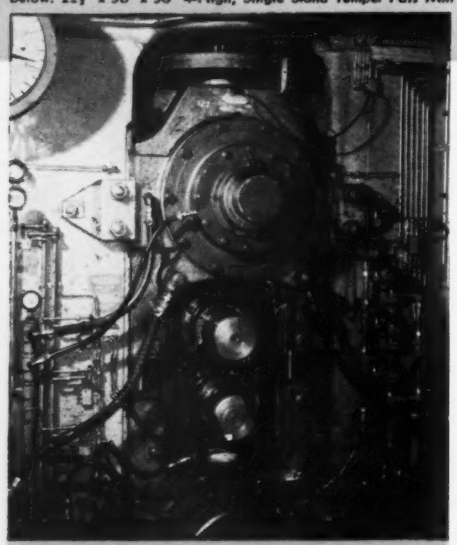
Electrofilm, Inc., of North Hollywood, California, has released a new solid film lubricant (22-T) designed for metal spinning, die stamping and forming, tube bending and wire drawing. This new solid film lubricant was developed to reduce damage and wear to parts and dies. Its low coefficient of friction also allows smaller radius bends on tubing. 22-T is applied either by dipping or spraying and can be washed off the finished part in 140° F water.

Circle No. 7 on Reader Service Card

(Continued on page 559)

MORGOIL
BEARINGS

Above: 21" x 56" x 56" 4-High, 4-Stand Tandem Cold Strip Mill
Below: 22½" x 56" x 56" 4-High, Single Stand Temper Pass Mill



MORGAN WORCESTER

MB 38

Meet the DEMAND of TOP QUALITY PRODUCTION

The demonstrated ability to meet high speeds, heavy loads and close tolerances made MORGOILS a natural choice for the back-up rolls on Inland Steel's new Cold Mills at the Indiana Harbor works.

Both the new 4-High, 4-Stand Tandem Cold Strip Mill and the new 4-High Single Stand Temper Pass Mill are equipped with 44"-90 MORGOIL BEARINGS.

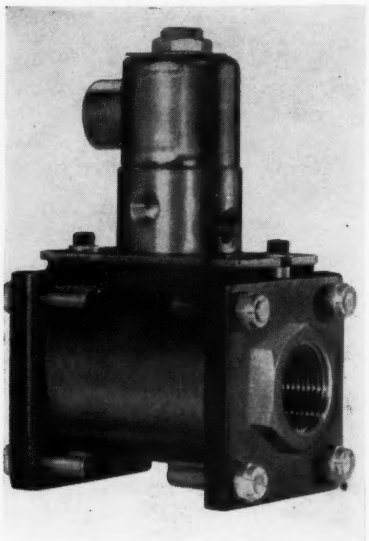
MORGAN CONSTRUCTION COMPANY
WORCESTER, MASSACHUSETTS
ROLLING MILLS • MORGOIL BEARINGS • WIRE MILLS
GAS PRODUCERS • EJECTORS • REGENERATIVE FURNACE CONTROL

NEW PRODUCTS

(*Continued from page* 557)

NEW AIRMATIC TUBE-O-MATIC VALVE MANUFACTURED

Airmatic Valve, Inc. announces a new Tube-O-Matic valve with a squeeze action which automatically controls the flow of air, oil, chemicals and even grinding slurries, without the use of poppets or spools. There are no metal to metal contacts and no metal to rubber seals.



When the valve is open, supply pressure through the inlet port forces an elastic rubber sleeve against the tube, forcing the media to flow around the inlet plug and through the outlet port. To close the valve, pilot pressure is applied between the inside diameter of the tube and the outer periphery of the rubber sleeve. This collapses the sleeve over the rounded ends of the inlet and outlet end plugs, sealing off the supply pressure. The only action is the flexing of the sleeve.

Pressures up to 200 psi can be handled. Pipe sizes: ¼" to 12". The valve is available either normally open or normally closed.

Circle No. 8 on Reader Service Card

SPECIAL SPROCKETS, GEARS, AND PINIONS AVAILABLE

Sprockets, gears, and pinions are available to customer specifications, flame-cut by a natural-gas-and-oxygen process which provides close tolerances and surface finishes. Reliable Steel Plate Co., Cleveland, Ohio, has developed the process to provide custom gears in production quantities at less than the cost of castings.

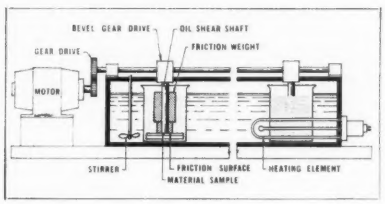
The natural-gas process employs a relatively cool flame and eliminates most of the surface hardening characteristic of flame-cut shapes and reduces machining costs by permitting higher cutting speeds and longer tool life.

The process can provide special gears of any design in quantities from one to thousands without pattern, template, or tooling changes. Thicknesses can be cut up to 28 inches without sacrifice of surface finish. Materials include low and high-carbon steel, manganese and silicon steels, high-carbon nickel, nickel-chromium, and molybdenum steels. The minimum practical thickness is ¼ in.; tolerance at this thickness is +.030.

Circle No. 9 on Reader Service Card

NEW OIL TESTER DUPLICATES ACTUAL OPERATING CONDITIONS

The Permawick Electric Motor Oil Oxidation Tester is a system developed to enable bearing designers and lubrication engineers to test up to six samples of oil simultaneously under laboratory conditions which are virtually indistinguishable from those encountered in actual bearing applications.



The new system, known as PEMOTT, has been developed by the Permawick Company, of Detroit.

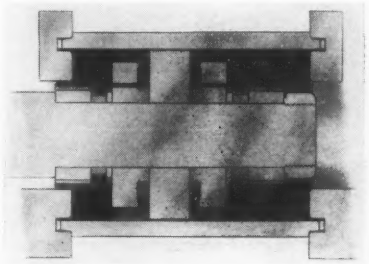
PEMOTT is claimed to be able to produce under controlled conditions the four physical phenomena which cause degradation in lubricating oils: 1) Continual shearing of oil 2) Heat, 3) Reaction on lubricant by ambient conditions, 4) Chemical or catalytic effects of metal wear debris.

Oil samples aged in the PEMOTT are then tested by standard ASTM methods to measure these three indices of degradation: 1) Changes in oil viscosity, 2) Increases in oil acidity, 3) Formation of insolubles.

Circle No. 10 on Reader Service Card

HIGH SPEED CUSHIONS FOR HYDRAULIC CYLINDERS OFFERED

A new, high speed, self-regulating, "floating" cushion for hydraulic cylinders is available throughout the complete line (1½" through 14" bores) of hydraulic cylinders manufactured by the Miller Fluid Power Division, Flick-Reedy Corporation, Bensenville, Illinois.



In high speed cycling, the floating cushion plunger acts as an extra large check valve that allows comparatively large volumes of fluid to enter the cylinder around the piston rod. The ball check assembly is replaced with a second adjusting screw which, together with the regular cushion adjusting screw, permits easy access for cushion adjustment with either of the two screws.

The cushion plunger has a thin-wall, floating sleeve which automatically adjusts itself to trapped pressure under varying deceleration conditions.

Circle No. 11 on Reader Service Card

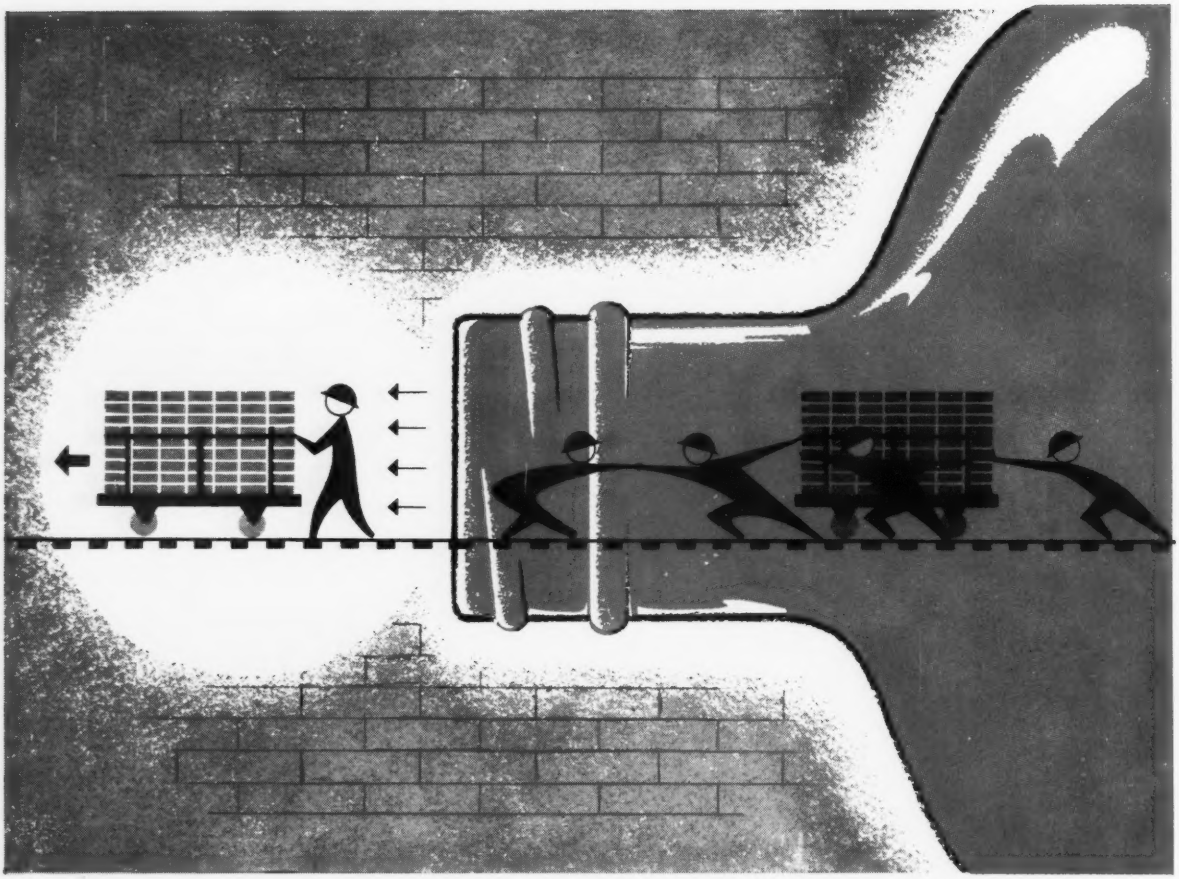
TWO NEW LINES OF STAINLESS STEEL PUMPS ANNOUNCED

Two types of stainless steel pumps are now being produced by Bell and Gossett Co. All wetted parts of the pumps in both new lines are made of 316 stainless steel. Many models are available in Uni-built design while others are of long coupled design. Capacities range up to 220 gallons per minute and heads to 120 feet. The seals are of "Remite" and are designed to be self-lubricating, leakless and virtually wear-proof.

These pumps are equipped with deep grooved ball bearings and standard flexible couplings. Alignment is said to be positive because the bearing, shaft, and impeller are entirely self-supported on one casting. In both pump series, the impellers are dynamically balanced to prevent shaft vibration and to extend seal life.

Circle No. 12 on Reader Service Card

(*Continued on page* 561)



New "hot" grease ends brick-oven bottlenecks

. . . and one man outperforms four!

Ceramic Combustion & Engineering Company, consultants for the Davidson Brick Co., Los Angeles, California, ran into production bottlenecks while using ordinary grease in dryer-car bearings. Oven temperatures (350° to 400° F.) cooked the grease solid . . . stalled the cars loaded with bricks in the dryer. Bearings seized so badly that four men were needed to move each carload.

Now—with Shell Darina Grease 2 as the lubricant, Davidson reports: "Bearings roll free, even at 400° F., one man, instead of the four previously required, can *easily* roll a carload of bricks . . . no more bottlenecks caused by frozen bearings."

Darina® Grease 2 is an economical, multi-purpose grease with exceptional stability in high-temperature applications. It offers these outstanding qualities:

HIGH-TEMPERATURE PERFORMANCE. *100° F. better* than conventional multi-purpose greases.

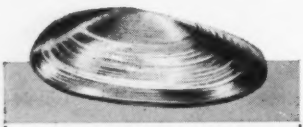
EFFICIENT SEALING. Contains no soap—nothing to melt and run out.

HIGH RESISTANCE TO WATER WASH-OUT. Doesn't dissolve or separate when mixed with water . . . lubricates efficiently under wet conditions.

POSITIVE RUST PROTECTION. Protects bearings even under severe moisture conditions.

ECONOMICAL. Savings up to 35% in grease and labor costs alone are possible because of Darina's superior performance over a wide range of plant operating conditions.

For more information on Darina Grease, write: Shell Oil Company, 50 West 50th St., New York 20, N. Y., or 100 Bush St., San Francisco 6, California. In Canada: Shell Oil Company of Canada, Limited, 505 University Avenue, Toronto 2, Ontario.



An Interesting Fact! Every Shell Branded Industrial Lubricant is named for a sea shell. Shown here: Darina solenoides.

SHELL DARINA GREASE

the multi-purpose, high-temperature grease

SHELL

NEW PRODUCTS

(*Continued from page* 559)

NEW VANE PUMP SERIES DEVELOPED

Denison Engineering Division, American Brake Shoe Company, Columbus, Ohio, has announced the development of a new series of 2500 psi vane pumps, known as the TIC series. Designed for continuous operation at pressures up to 2500 psi and at speeds to 2500 rpm, these compact units are available in four models. They are rated at eight, 11, 14 and 17 gpm, rotating at 1,200 rpm and at zero psi.

The new TIC pumps employ the same balanced vane, floating cartridge construction of all Denison "T" series. They consist of four major components: a housing with a ball-bearing, shaft seal and inlet connection; a pumping cartridge of rotor, vanes, cam ring and a floating front port plate; shaft; and an end cap with a roller bearing assembly and an outlet connection. The shaft is supported by this bearing and the ball-bearing in the housing. This support feature allows use with high side loadings on the shaft.

Circle No. 13 on Reader Service Card

NEW MULTI-PURPOSE LUBRICANT PUMP ANNOUNCED

A "multi-purpose" lubricant pump designed for dispensing industrial lubricants by providing either low pressure volume delivery or high pressure delivery at up to 5,000 psi from the same compact, portable, manually operated unit, has been announced by Alemite.

The pump, designated 7138 VP Pump, is two pumps in one, with a lapped-fit high pressure piston directly connected to the main pump rod and handle. This piston operates within the volume pumping piston.

For high volume the high pressure piston is locked to the volume piston, and both pistons operate and provide volume delivery of one cubic inch per stroke.

High volume delivery is for gear housings, large bearings, filling of oil reservoirs, centralized lubrication systems, and elsewhere. The high pressure is for any bearing requiring pressure lubrication; up to 5,000 psi.

Circle No. 14 on Reader Service Card

MAGNETIC LINE STRAINERS AVAILABLE

A new Magnetic Line Strainer has been developed by Ripley Screen and Strainer Co. The strainers are said to meet JIC specifications for magnets in hydraulic systems and are suitable for use on all types of equipment, including: mining equipment, diesel engines, machinery lubrication systems, oil filtration and mobile construction equipment. They are said to be adaptable for existing systems where sump strainers cannot be used.

Concentrated two-way filtration comes from powerful, built-in circular Alnico magnets placed between removable filter disc screens at specific intervals. Screens remove non-magnetic matter and the rest of the unit traps magnetic particles. According to the manufacturer, the strainers are capable of trapping small particles from .003" to .005". When disassembled, iron and metallic filings drop off the disc covers for quick, easy cleaning.

Circle No. 15 on Reader Service Card

REDESIGNED HEAVY DUTY SIGHT FEED VALVES OFFERED

Heavy duty sight feed valves, style SFK, are offered by Oil-Rite Corp. for controlling the dispensing of a liquid volume and to visually indicate its flow in a sight chamber. The flow volume is adjusted by a needle valve. This line has been available in angle, cross, inverted angle, and straight patterns.

The manufacturer states that some of the improved features are: the orifice or port has been enlarged, which makes this new valve capable of taking care of higher flow rates; and a new exclusive swivel arrangement has been incorporated into most models, which allows viewing windows to be turned into any desirable direction.

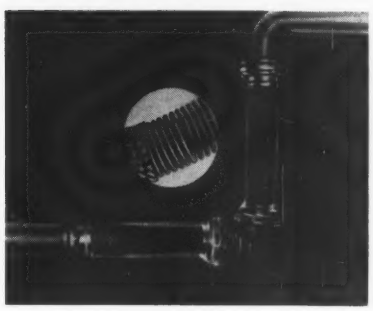
Heavy duty sight feed valves are said to have excellent flow characteristics with low pressure drop. The needle valve controls and regulates liquid flow from full flow to complete shut off.

Modifications are available upon request. These units are offered for use on gravity or pressure oiling systems for bearing lubrication where considerable capacities of oil must be dispensed.

Circle No. 16 on Reader Service Card

FLEXIBLE CONNECTORS FOR PUMPS AND PIPELINES INTRODUCED

Effects of pipeline vibration, misalignment, or thermal expansion in pump installations are claimed to be absorbed or reduced by new "H.A.P." Flexible Pump Connectors, introduced by Allied Metal Hose Company.



Consisting of a short section of corrugated flexible metal tubing, protected against pressure elongation by wire braid, these pump connectors are furnished with threaded or flanged end connections to fit all standard piping sizes.

Material, lengths, and construction of the connectors are varied to suit the conveyed fluid, pressures, temperatures, and movements encountered.

"H.A.P." Pump Connectors handle pressures to 1500 psig, depending on alloy, braid construction and size. Their helical, rather than annular, corrugations offer less resistance to flow, while distributing stresses along entire hose length to minimize localized failures. The safety factor of each hose unit is four times the rated working pressure.

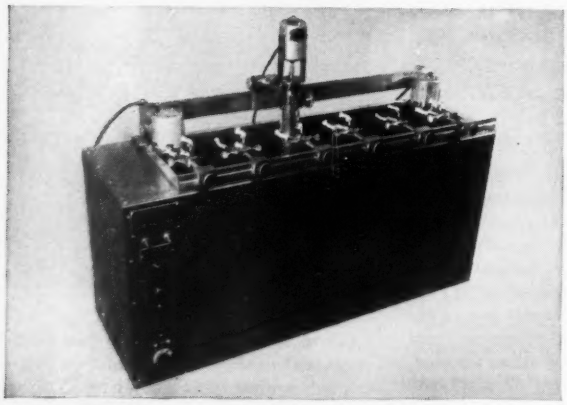
Circle No. 17 on Reader Service Card

If it rolls on an axle or turns in a bearing or rides on a shaft if it slides in a groove or moves on a pivot if it bores or cuts or transmits pressure one of Sinclair's 500 specialized lubricants is designed to make it work better. For answers to your lubrication problems, write today to

SINCLAIR REFINING COMPANY

Technical Service Division, 600 Fifth Ave., N.Y. 20 N.Y.

Circle No. 18 on Reader Service Card



New RAC unit yields demulsibility characteristics of lubricating oils

Utilizing the Wheeling Steel Test Method, this compact new RAC Demulsibility Bath accurately determines the three important factors in establishing lubricating oil characteristics: percentage of water in the oil, total ml. of "free water," and ml. of emulsion separated from the mixture.

The all stainless steel bath is well insulated to prevent heat losses, offers two continuous duty stirrers for even heat distribution and Hevi-duty electrical heaters that provide temperature range from 50° to 250°F. Available in six, four and two-sample units, bath has inside dimensions of 36" X 12" X 18" and is equipped with a variable speed emulsifier with maximum of 8000 RPM. Unit can be easily converted to a regular water bath for other test work. Write for FREE bulletin giving specifications and prices.

RESEARCH APPLIANCE COMPANY

Box 307 • Allison Park, Pa.

Circle No. 19 on Reader Service Card

Plan to attend

ASLE Annual Meeting

and

Lubrication Exhibit

at

Bellvue Stratford Hotel

Philadelphia, Pa.

April 11-12-13, 1961

PRODUCT LITERATURE

CATALOG ON REDUCERS ISSUED

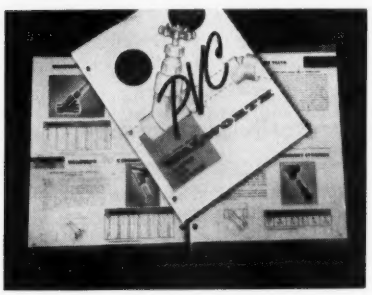
Application, selection and order information for the new line of "radicon" enclosed fan-cooled worm gear drives recently introduced by Foote Bros. Gear and Machine Corporation, Chicago, are detailed in a new "Engineering Catalog RCB."

The booklet is divided into three sections: an "Introduction" covers AGMA standards, in accordance with which all Radicon units are rated; outstanding design features of the new Radicon models; and general specification and sectional views. The second section, "Explanation-Ratings," details application data, selection and ordering procedures, AGMA load classifications and service factors, rating tables, and permissible overhung and thrust loads. Section three, "Descriptions-Dimensions," provides details and measurements of the several model types available.

Circle No. 20 on Reader Service Card

PLASTIC VALVE CIRCULAR PUBLISHED

A new circular describing plastic valves and fittings has been published recently by the Walworth Company. This sixteen-page folder describes in detail the various Walworth products available in corrosion-resistant polyvinyl chloride (PVC).



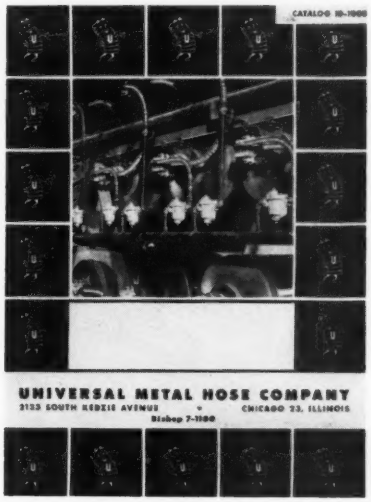
One feature of the new 3-color piece is a comprehensive, 4-page Application Table that rates PVC's effectiveness on over 300 fluids. Another section gives illustrated descriptions and dimensions of all Walworth's PVC valves (Y-globe, check, bleeder and diaphragm), strainers, unions, fittings and expansion joints.

Copies of the new circular are available upon request.

Circle No. 21 on Reader Service Card

ALL-METAL FLEXIBLE HOSE FEATURED IN NEW CATALOG

A new 20-page catalog, No. ID-100D, describing Universal All-Metal Flexible Hose to convey gases, solids and fluids; for vibration control; and to compensate for motion or misalignment under high temperature, pressure, vacuum or abrasive conditions, has been released by Universal Metal Hose Company.



Detailed descriptions, specifications, and applications-use together with coupling recommendations, are given for the Universal U-200 and U-250 Series for conveying searching gases and fluids like steam, hot oil, gasoline, liquefied gases, refrigerants, chemicals, etc., where completely leak-proof hose construction is essential.

Circle No. 22 on Reader Service Card

NEW WALL CHART FOR REPLACEMENT HYDRAULIC HOSE, COUPLINGS AND FITTINGS OFFERED

Users of fluid power will find the completely new 21" x 32" two-color Wall Chart just released by Anchor Coupling Co., Inc., a quick, "ready-reference" for all Anchor Reusable Hydraulic Hose, Couplings and Fittings.

The new Anchor Wall Chart, with illustrations of all Anchor Reusable Hydraulic Hose, Couplings and Fittings, showing dimensions, enables fluid power users to easily find the Anchor catalog numbers of the Anchor Reusable products shown.

In addition, Anchor has released a new handy 10-page, envelope-size leaflet giving highlights of their complete hydraulic hose, couplings and fittings for replacement use.

Both the new Wall Chart and leaflet tie in with the 24-page Anchor Distributor Catalog No. D-1 and Supplement covering Anchor Pressed-On Hydraulic Hose Assemblies and Reusable Hydraulic Hose, Couplings and Fittings, which give detailed data, specifications and complete instructions covering Anchor replacement products for on-the-job installation.

Circle No. 23 on Reader Service Card

CATALOG ON CYLINDER, DRILL UNIT AND VALVE LINE PUBLISHED

A new 2-color 24-page catalog is available from the Alkon Products Corporation on its complete line of cylinders, drill units and valves. This catalog has been designed to provide an easy-to-read, single reference source on the complete line of pneumatic and hydraulic components.

Included in the material is product data and specifications, schematics, price lists and discount schedules. An added feature is the introduction of the new Series B inter-changeable cylinder for air and oil service.

Circle No. 24 on Reader Service Card

We are pleased to announce that The Alpha Molykote Corporation, Stamford, Connecticut, has become an Industrial Member of the Society. Mr. Alfred Sonntag, President, will be the industrial representative.

NEW ASLE MEMBERS OCTOBER, 1960

PHILADELPHIA

Joseph E. Thompson
Atlantic Refining Co.
Philadelphia, Pennsylvania

CONNECTICUT

Clyde R. Miller
The G. H. Berlin Oil Co.
Hartford, Connecticut

TORONTO

Don J. Coward
Imperial Oil Limited
Downsview, Ont., Canada

George R. Burns
Sun Oil Co., Limited
Toronto, Ontario, Canada

NORTHERN CALIFORNIA

Wolfgang A. Kempff
Nat'l. Seal Div. Federal
Mogul Bower Bearing Co.
Redwood City, California

BOSTON

P. Joseph LeMaistre
Texaco, Incorporated
Chestnut Hill, Massachusetts

William W. Ware
New Hampshire Ball Bearings, Inc.
Peterborough, New Hampshire

CHICAGO

Arnold J. Parus
Standard Oil Co. (Ind.)
Chicago, Illinois

E. R. Pawlisch
Refinery Products Co.
Schiller Park, Illinois

LOS ANGELES

Paul L. Polizzotto (R)*
E. F. Houghton & Company
Los Angeles, California

Donald A. Swerdfiger
AiResearch Ind. Div.
Los Angeles, California

At the close of the Joint Lubrication Conference which was held in Boston, the Boston Section held a meeting at which S. R. Calish, Jr., California Research Corporation, presented an interesting talk before a packed house on "Automotive Gear Lubricants." Several national officers and directors were present including: J. Boase, Canadian Regional Vice President; L. E. Hoyer, Director; C. L. Willey, Executive Secretary; "Red" Giard, Chairman, Boston Section; and the speaker, S. R. Calish, Jr., also a Director. Photo courtesy of George Kellis, Area Representative, Eastern Region.

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SECTION NEWS

Montreal—ASLE President, L. O. Witzenburg (Cleveland Worm and Gear Division) addressed the December 7 regular meeting. His subject was "Worm Gear Lubrication."

Code 2467—Section No. 41—That's the simple administrative procedure necessary to register OKLAHOMA as the 41st ASLE Section. Behind all of this is a lot of hard work, spear-headed by Bill Mitacek, Phillips Petroleum Company, Bartlesville, Oklahoma. The petition to establish this section was approved on October 18 by the Board of Directors. The first formal meeting of the section was held during November at which time the charter was presented and section officers were elected.

Baltimore—The Section's calendar of events reveals that on November 17, 1960, Mr. W. Schreiber (Supervisor, Additive Research—The Atlantic Refining Co.) was guest speaker. His subject was "Additives—Facts or Fiction."

Hudson-Mohawk—A schedule of the programs for the 1960-1961 term shows that Dr. W. E. Campbell, Consulting Chemist, will speak at the January 19, 1961 meeting. His topic will be "Solid Lubricants in Industry." The Hudson-Mohawk Section is making arrangements to have a Mobile Lubrication Training Unit spend a day in the Schenectady area during one of the spring months.

Indianapolis—The agenda of the Steering Committee meeting on October 10 included the election, by acclamation, the following officers: Chairman—Kenneth Carvan (Haynes Stellite Co.), Vice Chairman—Harold McCoy (Delco Remy Div., GMAC), Secretary-Treasurer—Robert F. Kraft (Harkins Engineering Sales Co.), Program Chairman — Donald Wilson (Continental Steel Corp.).
The Section's first meeting of the Fall Season was held on November 1. At that time, William M. Robinson (Master Mechanic, Steel Division, Continental Steel Corp.) was the guest speaker.

Mr. Louis M. Fleming (left) is shown receiving a plaque commemorating his work as Chairman of the Twin Cities Section of ASLE during the 1959-1960 period from Mr. Donald J. Armstrong. Mr. Fleming is associated with Cities Service Oil Company and Mr. Armstrong with Farmers Union Central Exchange. The presentation was made at the first fall meeting of the ASLE on Thursday, September 29, 1960 at Minneapolis, Minnesota.

Houston—Mr. Gayle A. Chaffin, who was elected as First Vice Chairman of the Houston Section, found it necessary to resign due to a change in his company responsibilities. Mr. Gilbert K. Ludwig (Continental Oil Co., Houston, Texas) has been nominated to take Mr. Chaffin's place.

New York—Mr. Bruce Dunham, Technical Consultant in Sun Oil Company's Industrial Products Department, was guest speaker at the November Section Meeting. Mr. Dunham discussed the various types of hydraulic fluids currently used to accommodate the variety of operating conditions experienced in industrial plants. He emphasized modernization of lubricants to accommodate the improvements in hydraulic design.

Buffalo—The subject of the November meeting was "A Discussion of Present Practices in Lubrication Design by Machinery Builders and How These Affect Lubrication Practices in Industrial Plants and How Practices in Plants Affect Consideration in Lubrication Design on the Part of the Machinery Builders." The speaker was Mr. Dalton Schnack, Lubrication Engineer for the Equipment Builder Division of Socony Mobil, Pittsburgh, Pa.

Saginaw Valley—Ladies' Night will be observed at the regular monthly meeting on January 26. John W. Mestrezat (J. N. Fauver Co.) is scheduled to be the guest speaker.

Chicago—The regular monthly meeting on January 19 has been designated as Bosses' Night and National Officers' Night. Mr. C. E. Genovese, U. S. Steel Corp., Chicago, will be the guest speaker. His subject will be "Management Looks at Lubrication."

Rochester—The subject, "Let's Stop Industrial Rust," was discussed at the November meeting by Mr. James T. DiPlazza, Senior Sales Engineer, Esso Standard Division.

Boston—The program for the January meeting will feature Mr. L. O. Witzenburg, ASLE President.

Vol. 16, No. 12

Lubrication Engineering

did you cast your vote

. . . for the *subjects* of your choice?

Just a few weeks ago we deposited in the mail a letter and a questionnaire from the Editor of *Lubrication Engineering* requesting that you cast your vote for subjects to be covered in the months ahead. Did you, as a shareholder in this publication, exercise this privilege *and responsibility?*

Editing a technical publication reaching into many industries and encompassing many functions such as sales, service, consumer, research, education, and design, requires help—only the kind of help that you as a reader of *Lubrication Engineering* can provide. This is where you can, and should, exercise your responsibility.

In introducing the series of articles on "The Lubrication Engineer in Action", which has been well received, it was stated that "to know ourselves" is ASLE's greatest need today. This is another step in the direction of knowing ourselves—by knowing what our readers want! The practical plant-type articles that have been featured during the past months are only a beginning—not an end. The continuation of this program into 1961 appears to be most desirable, and with your help and your vote we can set our course in the right direction.

There are many subjects that can be treated in these pages, and we don't presume to have listed them all on the survey form. Add those that we have missed, delete those that appear unimportant, change those that need change. The point is—be positive—express your thoughts—cast your vote. *Lubrication Engineering* is *your* publication, help us to make sure it suits your needs.

C. L. Willey
ASLE Executive Secretary

The Lubrication Engineer in Action

Report No. 6 — At the Verson Allsteel Press Company in Chicago.

The previous reports in this series have been based on the achievements of individual lubrication engineers as they solved the problems encountered in specific production operations. In total, the practices and methods these men have evolved comprise a practical foundation for the general science of lubrication engineering.

This report deals with the gains that can be made in typical manufacturing plants of all types through application of general lubrication engineering. Specifically, it is the story of how they have made good lubrication engineering practices a part of everyday routine at Verson . . . and underscores the opportunities for improvement through properly planned and programmed lubrication that exist in any plant.

Partial view of the Tooling Division where Verson press dies are produced.

Good lubrication is easy to believe in. Who, for instance, will doubt that, "An ounce of oil in time saves 9 hours of repair work," or, "A pound of grease is worth a thousand hours of bearing life."

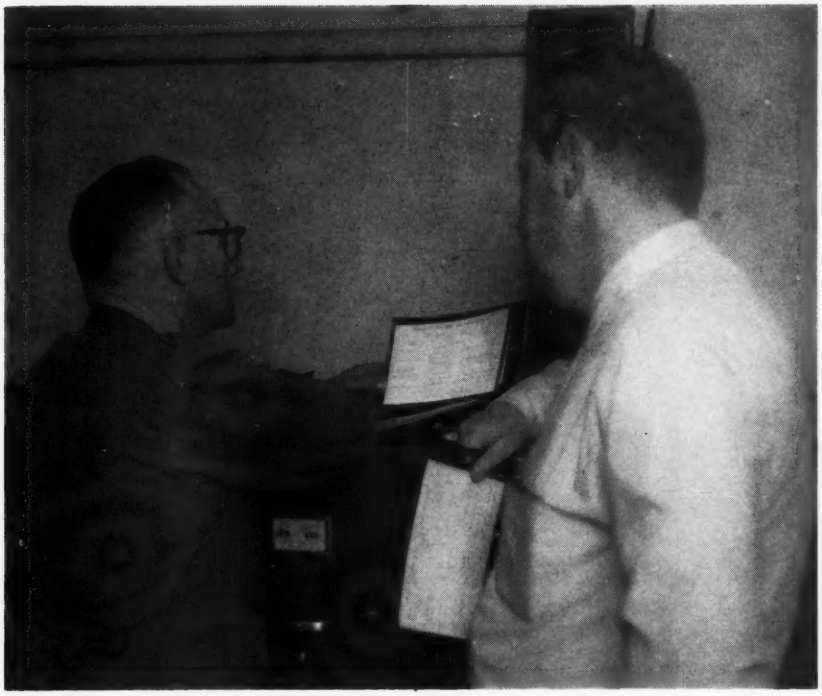
Fortunately, good lubrication is also just as easy to practice as it is to believe in—far easier than the widely accepted truisms just paraphrased. This story of how it's done at Verson points the way. And the success formula they use will work wherever a shaft turns in a bearing—it's as simple as A, B, C:

A. Recognize the need for special attention to lubrication requirements.

B. Be sure that "special" attention is also competent, correct attention.

C. Be consistent.

Recognizing the need for lubrication engineering is more a matter of initiative than of discovery. As pointed out earlier, everyone in industry readily agrees to the importance of good lubrication. *But within the*



Earl Lewis (left), mechanical maintenance foreman, and Vito Budrionis, chief plant engineer, inspect lubrication card file. The file cards outline lubrication procedures on equipment.

management and supervisory organization of a plant, the need for lubrication engineering can not be considered recognized until some one either assumes or is given the responsibility for making sure that the area is properly covered.

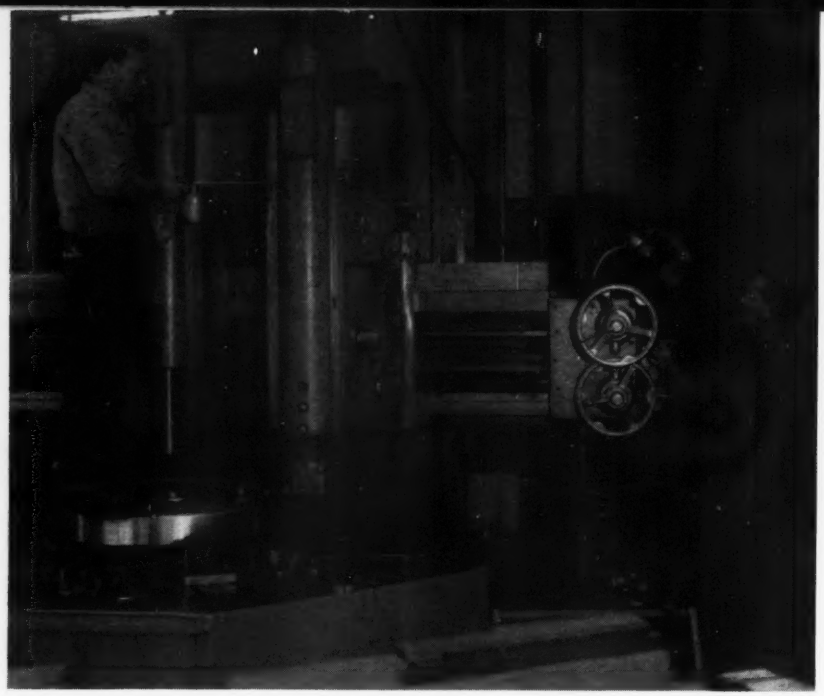
To recognize the need for lubrication engineering requires that a decision be made to do something about it.

In making sure that the right things are done, good authority and counsel is easily available to everyone. In addition to wide coverage in the technical press and detailed literature on almost every conceivable type of problem, just two sources provide a virtually complete and correct guide to plant lubrication. These are (1) the manufacturers of the equipment to be lubricated and (2) the suppliers of the lubricating materials to be used. Many suppliers of lubricating materials and equipment actually maintain a staff of lubrication engineers for the sole purpose of rendering assistance to users.

Consistency depends on the establishment of some routine or system that will automatically and continuously assure execution of good lubrication practices. Actually, this last of the "three commandments" of good lubrication engineering is the biggest pitfall.

The need for improved lubrication practices is almost certain to be clearly recognized at times in every plant, as, for instance, when a costly piece of equipment breaks down and someone finally confesses to the master mechanic that the main bearings had been running dry for the last three shifts.

Lubrication cart used in plant by lubricating crew.



The lubricating crew, Leonard Verrilli and Frank Mistro performing routine lubrication service on a machine tool in the gear department.

In cases like this, recognition of the need is definite and often unpleasantly dramatic . . . and inevitably followed by a flurry of zealous attention to proper procedure.

But then, all too soon, the uproar is forgotten and lubrication engineering activity is allowed to subside to the "let George take care of it" stage again. And the program flounders on the rocks of inconsistency until the next serious emergency.

Consistency of good lubrication practice is the only way to guard against the costly breakdowns that are the certain aftermath of oil and grease errors.

It is immediately apparent that the best assurance of adherence to the three commandments of lubrication engineering is a lubrication engineer on the staff. And this is the way it's done in most large operations or where the lubrication problems are unusually complex and critical. Recognition of the need by someone who will do something about it as well as competency and consistency of practice are built-in.

But where staff functions are not so finely divided or specialized (and where, by the same token, the problems are not likely to be so complex), good results are still certain if only the three simple commandments outlined above are followed. Alert, conscientious management proves it at Verson in Chicago.

The plant covers about 22 acres. A complement numbering roughly 900 employees operates light and heavy metal working machinery of all kinds in turning out the presses and associated metal forming equipment that have made Verson one of the well known and respected names in industry. In addition to the usual array of material handling and utility equipment, between 450 and 500 pieces of production machinery are in operation.

The magnitude of their lubrication problem is indicated by annual consumption of approximately 20,000 gallons of lubricating oils and cutting fluids and 20,000 pounds of grease. How well these problems are solved is indicated by the fact that their present procedures have been in effect for more than 10 years without significant change.

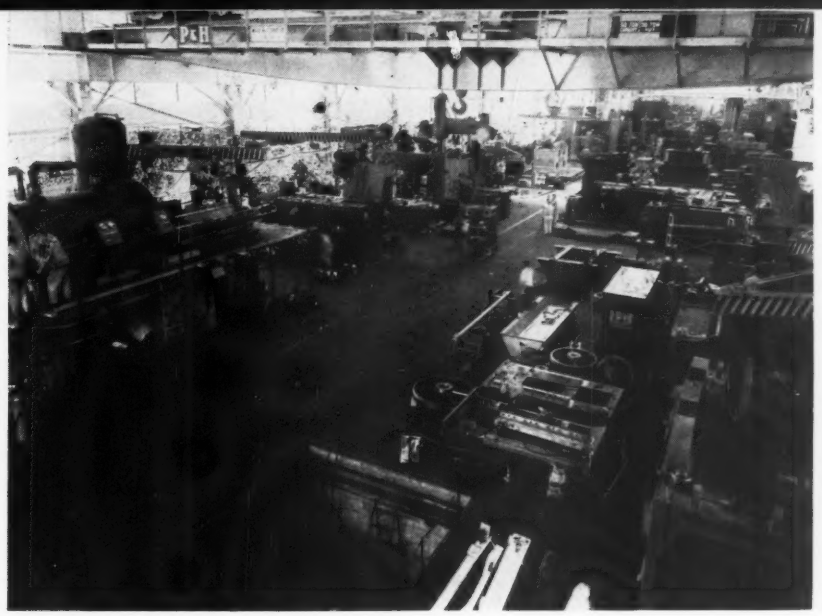
The prime movers behind good lubrication engineering at Verson, the men most concerned and who make sure that something is done, are Vito Budrionis, chief plant engineer, and Earl Lewis, mechanical maintenance foreman. In their operations, lubrication is the key factor in a successful preventive maintenance program.

Behind Verson's vigilance in recognizing the importance of good lubrication engineering is a well formulated estimate that equipment maintenance costs would go up 35 to 40 per cent immediately if the system were relaxed! The system itself, primarily designed to comply with equipment manufacturers' lubrication recommendations and based on close technical cooperation with the company's principal lubricant supplier, was started just after World War II.

Mainstay of the system in assuring consistency is a special check-off card file. For each machine or piece of equipment requiring lubrication, a separate card in this file calls out the proper lubrication procedure. Spaces are also provided for entering a running log of lubrication service.

The card file is maintained by a 2-man lubrication service crew that covers all areas of the plant on an eight day cycle. Daily lubrication requirements are, as a general rule, quite simple and not too critical. These are left to the individual machine operators at Verson.

The men of the lubrication service team are both well trained specialists, familiar with the needs and characteristics of all of the equipment entrusted to their care. A good example of their alert performance occurred when a vital oil line broke in one of the company's big grinding machines.



A portion of the main assembly area where presses are completed for shipment. A number of the large machine tools used in the manufacture of this equipment can be seen in the background.

The leak was internal and would not have been detected in time to prevent serious damage if it had not been for the oiling crew. They noticed the unusual volume of make-up oil required in the course of their routine servicing and traced the fault. Repair was simple, cost and lost time were minimum.

The only units not serviced by these specialists are the anti-friction bearing electric motors. These are made the specific responsibility of the electricians as a part of their own routine maintenance work. Sleeve bearing motors are also separately handled but are serviced by the regular lubricating crew.

Incidentally, one man handled the regular lubricating chores until just a few years ago. The second man was added when the first man could no longer cover the assignment. Significantly, bearing failure difficulties dropped off by 75 per cent as soon as the lubricating crew was doubled.

From this, it might be concluded that even more men would produce still further gains. On the contrary, the system is operating so well now that dramatic improvement would be impossible. Also, increasing application of automatic lubricating devices and equipment to machines of all types is having the effect of holding lubrication manpower needs constant as the plant grows.

The role played by Verson's principal lubricant supplier in the development and maintenance of this system is a big one. The card file itself, with its detailed instructions on proper lubrication procedure, was originally prepared with the close cooperation of the supplier's technical staff. As new equipment is put into use from time to time, the new cards necessary to keep the file complete are also prepared by the supplier.

In addition to this constant and detailed technical support, via the card file, the lubricant supplier also prepares a periodic plant-wide review or survey of all lubrication procedures in use. This keeps the system up to date and provides for reconsideration of established practices against new and possibly better procedures.

Even special or emergency problems fit into this pattern of technical cooperation between supplier and user. A case in point was a large machine that had to be taken out of service every six months to permit scraping of its cast iron ways. Analysis of this costly and troublesome situation resulted in the installation of composition ways and increased lubricant pumping capacity. The machine has now been in service for more than six years without interruption.

Another important supplier service, essential in the long term up-grading of lubrication on a plant wide basis, is cooperation and assistance in personnel training. Special educational films and manuals are made available and training clinics are led by supplier technicians to acquaint as many Verson employees as possible with the importance of good lubrication and proper procedures.

The results achieved under this system of alert, consistent action based on carefully planned procedure put Verson well up in the top bracket of industry for lubrication engineering efficiency. Application of the newest and most modern materials or methods is assured. Procurement costs are held to a minimum and stocking problems are simplified—only 7 different oils and 3 different greases meet all requirements. And, most important, overall maintenance costs are greatly reduced by substituting prevention (via good lubrication) for repair.

Actually, the success of the entire lubrication setup at Verson is best indicated by the smoothness of its operation and the lack of management involvement in what could otherwise be serious problems.

Lubrication engineering is a powerful force for greater operating efficiency—readily available to all and easy to apply. In some ways it can be almost as miraculously beneficial as the wonderful genie that Aladdin conjured up by rubbing a lamp. Remember, it was an oil lamp!

The Relationship Between the Oil-Water Ratio and the Effectiveness of Inhibitors in Oil Soluble Emulsions

by V. Carlson and E. O. Bennett,
Department of Biology, University of Houston, Houston, Texas

Nineteen antibacterial agents were studied in varying oil-water ratios of the same soluble oil emulsion-type metal cutting fluid. It was noted that the oil-water ratio can have a profound effect on the antimicrobial activity of these inhibitors and that a 1-40 dilution is the most difficult to inhibit. The amount of compound required to inhibit rises sharply as dilutions are increased from 1-10 to 1-40. This amount decreases slightly as the dilutions approach 1-70 and then fall off in a 1-90 dilution.

Only one compound, 1-bromo-1-nitropropane, was effective in the same concentration in all oil-water ratios while 2,4,6-trichlorophenol was most markedly affected by differing oil-water ratios. Three outstanding nitroparaffin derivatives, 2-nitro-2-methyl-1,3-propanediol, 2-nitro-2-ethyl-1, 3-propanediol, and 2-nitro-2-ethyl-1,3-propanediol dipropionate, were demonstrated in this investigation but in general, sodium ethyl mercuri thiosalicylate appeared to be the most effective inhibitor studied.

During the past few years, many publications have appeared concerning the use of chemical agents to control microbial growth in soluble oil emulsions, (1), (4), (7), (5). These studies have resulted in the discovery of two outstanding inhibitors that can be used in these products. Wheeler (6) found that sodium ethyl mercuri thiosalicylate, and Wheeler and Bennett (7) observed that tris (hydroxymethyl) nitromethane were outstanding antibacterial agents against the microbial flora found in emulsions of metal cutting fluids. An experiment in this laboratory showed that sodium ethyl mercuri thiosalicylate in concentration of 500 ppm was still completely inhibitory in an emulsion after 910 days exposure to the atmosphere. This compound retains its inhibitory activity longer than the average life of the emulsion under normal operating conditions.

The effectiveness of any inhibitor in a given situation is dependent upon a number of factors. Bennett *et al.* (2) and Bennett and Futch (3) have shown that bacteria exhibit variable sensitivities to the same inhibitor in emulsions of different soluble oils. For example, it required from 50 to 150 ppm of 2-nitro-1-butanol to inhibit the same culture of *Pseudomonas aeruginosa* in six different soluble oil emulsions. In similar experiments, it was shown that sodium ethyl mercuri thiosalicylate was the only compound studied that did not show this variation from product to product. Pseudomonads were consistently sensitive to 10 ppm of this inhibitor in all six emulsions.

The oil-water ratio is an additional factor that can govern the effectiveness of antimicrobial agents in soluble oil emulsions. The importance of this factor will be shown in this paper. Nitroparaffin derivatives, sodium ethyl mercuri thiosalicylate, and 2,4,6-trichlorophenol were employed as inhibitors in this study.

EXPERIMENTAL PROCEDURE

A pure culture of *Pseudomonas aeruginosa* was used throughout this study. The culture was grown in trypticase-glucose broth for 24 hours at 37 C. The cells were harvested by centrifugation, washed once in 0.87% saline, and resuspended in the same concentration of saline under aseptic conditions. This suspension was standardized to an optical density of 0.1 using a photoelectric colorimeter with a blue filter.

The soluble oil used in this study contained no inhibitor. The product was studied in different ratios varying from one part oil and nine parts water to one part oil and ninety-nine parts water. The inhibitors were added to the product in varying amounts as indicated in the table.

The inhibited emulsion was dispensed in 10 ml quantities into plugged test tubes. The tubes were autoclaved at 121.5 C at 15 pounds steam pressure for 15 minutes. After the tubes cooled, they were inoculated with 0.1 ml of the standardized cell suspension.

The inoculated tubes were incubated at 30 C for 48 hours, and then subcultures were made by inoculating a portion of the emulsion samples into nutrient broth. The nutrient broth tubes were incubated at 37 C for 48 hours and examined for the presence or absence of growth. The data shown in the tables are based upon growth in the subcultures.

The experiments were done in duplicate and appropriate controls were included in each experiment.

RESULTS

The results show that the oil-water ratio may have a profound effect upon the inhibitory activity of antibacterial agents in soluble oil emulsions. Statistical analysis of all the data indicated that the microbial flora of a 1-40 dilution is the most difficult to inhibit (Fig. 1). The quantity of compound required to inhibit rises sharply as increasing dilutions are made from 1-10 to 1-40. The curve then drops slightly to a 1-70 dilution and then falls off to a 1-90 dilution. There is no significant difference in the amount of inhibitor required in a 1-90 or 1-100 dilution.

It may be observed in Table 1 that the oil-water ratios have the most profound effect on 2,4,6-trichlorophenol. Only 50 ppm were required in a 1-20 dilution while 1100 ppm were required in a 1-40 dilution.

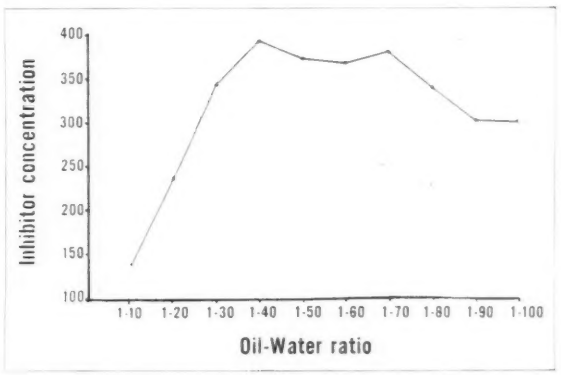


Fig. 1. Statistical analysis of the effects of the oil-water ratio on the activities of inhibitors in an emulsion oil.

TABLE 1

THE EFFECT OF THE OIL-WATER RATIO ON INHIBITORY ACTIVITY

(Results are expressed as inhibitory concentrations in ppm)

Inhibitor	Oil-Water Ratio									
	1-10	1-20	1-30	1-40	1-50	1-60	1-70	1-80	1-90	1-100
2-Nitro-2-ethyl-1, 3-propanediol	20	30	30	40	50	50	50	40	40	40
1-Bromo-1-nitropropane	100	100	100	100	100	100	100	100	100	100
2-Bromo-2-nitropropyl acetate	250	250	250	250	250	250	100	100	100	100
2-Nitro-1-butyl acetate	50	100	100	100	100	100	100	100	100	100
2-Nitro-1-propyl acetate	325	350	375	400	400	375	375	350	350	350
2-Amino-1-butanol	100	250	300	300	300	350	350	350	350	350
2-Nitro-2-methyl-1, 3-propanediol	40	40	40	50	50	40	40	40	40	30
2-Nitro-1-butanol	90	90	90	150	100	90	90	90	90	90
2-Methoxy-3-nitropentane	50	250	500	750	500	500	500	500	500	500
N-(2-nitroiosobutyl)-isopropylamine	50	50	100	100	100	100	100	100	100	100
2-Nitro-1-phenyl-1-propene	250	750	750	750	1000	1000	750	750	750	750
2-Nitro-2-ethyl-1, 3-propanediol dipropionate	50	50	50	100	50	50	50	50	50	50
1-Methoxy-2-nitro-1-phenylpropane	250	500	750	750	750	750	1000	750	750	750
1-Methoxy-2-nitro-2-bromo-1-phenylpropane	250	500	500	500	500	500	750	500	250	250
2-Nitro-1-p-chlorophenyl-1-propene	100	250	500	500	500	500	500	500	500	500
1-(2,4-dichlorophenyl)-2-nitro-1-propene	100	250	500	750	750	750	1000	750	500	500
1-p-Methoxyphenyl-2-nitro-1-propene	500	500	500	750	750	750	750	750	750	750
Sodium ethyl mercuri thiosalicylate	5	10	75	75	15	10	5	5	5	5
2,4,6-trichlorophenol	50	50	1000	1100	800	700	600	600	400	400

Only one compound, 1-bromo-1-nitropropane, was effective in the same concentration in all oil-water ratios.

It will be noted that, in general, sodium ethyl mercuri thiosalicylate was the most effective compound studied in the present investigation.

DISCUSSION

The oil-water ratio of a soluble oil emulsion has a profound effect on the antimicrobial activity of inhibitors in these products. It may be observed that changing the ratio can markedly increase the amount of inhibitor necessary for inhibition to a point where use of the particular compound is no longer practical.

The results point out the difficulties and pitfalls of adding an inhibitor to the concentrated product by the manufacturer. For example, if a manufacturer wanted to incorporate 2,4,6-trichlorophenol into his product, it would be necessary to add enough inhibitor so that there would be a minimum concentration of 1100 ppm if a 1-40 dilution is used. This would be almost three times the concentration needed for inhibition if the user employed a 1-100 ratio. It must be remembered that strain variation and type of soluble oil also plays a part in determining the final concentration of the inhibitor required. Bennett *et al.* (2) found that some strains of *Pseudomonas aeruginosa* require 2000 ppm of 2,4,6-trichlorophenol for inhibition. In addition, these workers found that sensitivity also depended upon the particular product used in the investigation.

Previous investigators have employed different oil-water ratios in their inhibitor studies. Pivnick and Fabian (4) used a 1-25 ratio in their screening test and a 1-50 ratio in their open system tests. Wheeler and Bennett (7) and Bennett (1) used a 1-40 ratio while Pivnick and Fotopoulos (5) used 1-10 and 1-100 oil-water ratios in their studies. These differences make it difficult to compare the results of the different investigations. The results of the present investigation indicate that a 1-40 oil-water ratio would be ideal for testing the effectiveness of most inhibitors in soluble oil emulsions.

It may be noted that 1-bromo-1-nitropropane was effective in the same concentration in all of the different oil-water ratios used. It is possible that in the future additional compounds will be found that exhibit the same type of activity. If these inhibitors are compatible with the formulation, they would be ideal antimicrobial agents for soluble oil emulsions.

There were three outstanding nitroparaffin derivatives demonstrated in the present investigation. These compounds (2-nitro-2-ethyl-1,3-propanediol, 2-nitro-2-ethyl-1,3-propanediol, and 2-nitro-2-ethyl-1,3-propanediol dipropionate) showed considerable activity in all of the oil-water ratios.

It may be noted that in general, sodium ethyl mercuri thiosalicylate was the most effective inhibitor studied in this investigation. This observation substantiates early work showing the superiority of this compound in controlling microbial growth in soluble oil emulsions.

SUMMARY

It has been shown that the oil-water ratio is an important factor in the effectiveness of antibacterial agents in soluble oil metal cutting fluids. The concentration of a specific antibacterial agent to inhibit *Pseudomonas aeruginosa* in an emulsion of a given fluid will generally vary with the oil-water ratio of the emulsion. It appears that for most antibacterial agents, the maximum concentration is required for an emulsion ratio of about 1-40, although this can vary depending upon the antibacterial agent and soluble oil involved.

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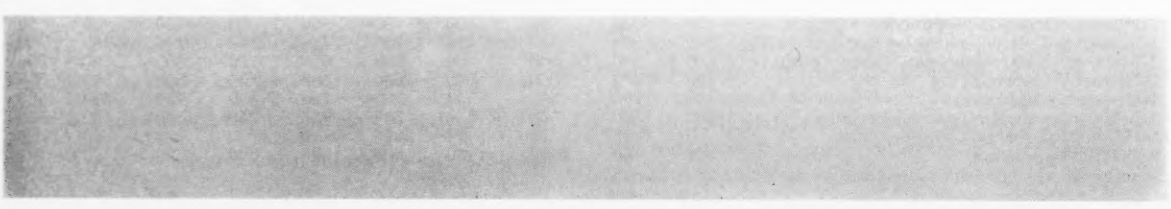

THE TRANSITION

From Boundary to Mixed Friction

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This paper presents the results of an investigation carried out to determine how various factors influence the surface velocity at which the transition from boundary to mixed friction occurs in flat sliding bearings. Friction measurements reveal that the velocity at which the transition occurs decreases with: (a) increasing base oil viscosity; (b) decreasing surface roughness; and (c) decreasing load. Except in cases where changes in the surface finish or bulk viscosity may result, the use of an additive does not affect the transition velocity. Moreover, calculations indicate that the percentage of the total load carried by hydrodynamic oil films under given conditions, is not affected by the use of an additive. The percentage does change with variations in load, surface finish and base oil viscosity. These calculations are made by comparing the magnitude of the coefficient of friction under the given conditions to the coefficients of boundary and hydrodynamic friction. It becomes evident that factors tending to cause an increase in the hydrodynamic load carrying ability, also tend to cause the transition from boundary to mixed friction to occur at a lower surface velocity.

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INTRODUCTION

The engineer's most valuable tool in the control of wear and friction is lubrication: In general, he attempts to separate two bodies in relative motion by a fluid film of lubricant. The fundamental principles of this type of lubrication have been well established since the pioneer work of Osborn Reynolds in 1886. Often, however, it is not possible or practical to maintain a complete fluid film. This results in more complex and less fully understood conditions.

While many investigators have added to our knowledge of the complex conditions, most of the work has been done with various types of apparatus other than flat sliding bearings. It is the purpose of this paper to present data of practical nature concerning one phase of partial film lubrication in flat sliding bearings.

Types of Friction

The friction occurring in the presence of lubrication can be divided into three types (1).

1. Boundary Friction

When bearings are exposed to the atmosphere, to a lubricant, or to any other form of contamination, the surfaces become covered by one or more films. The contamination may or may not be formed by a chemical reaction with the bearing material. Boundary friction results when the bearing surfaces are separated by some sort of material other than a fluid. Coefficients of boundary friction depend mainly upon the shear strength of these materials. It should be noted that these films are often ruptured so that boundary friction and intimate contact occur at different areas in the same bearing.

2. Hydrodynamic or Fluid Friction

As stated before, engineers usually strive to maintain a fluid film between bearing surfaces. The resulting hydrodynamic friction depends upon the thickness of the film, the viscosity of the film, and the relative velocity.

3. Mixed Friction

Frequently, bearings are operated under conditions which permit only a partial film so that boundary and hydrodynamic friction exist at different areas in the same bearing. This results in what is called mixed friction. Since the value of boundary friction is almost always higher than the value of fluid friction, mixed friction is characterized by a falling friction as the transformation from boundary to fluid friction takes place. Fig. 1 shows a typical curve in which the coefficient of friction is plotted against the velocity of a flat sliding bearings. This bearing is lubricated by a material that gives a boundary friction that is independent of sliding speed, as if often the case. It is possible by means of such a curve to determine the surface speed at which the transition from boundary to mixed friction takes place. Mixed friction conditions are often thought of as those conditions under which part of the load on a bearing is supported by areas of fluid films and partly by areas of boundary films.

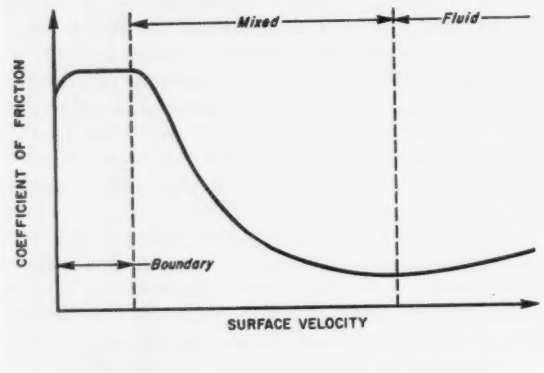

Fig. 1. Typical curve relating friction of sliding bearings to surface velocity. Shown are areas of boundary, mixed and fluid friction.

Development of Fluid Films in Flat Sliding Bearings

The classical theories of lubrication are limited to bearings that have converging wedges or other constrictions, such as those found in journal bearings and tilting-pad thrust bearings. Many texts are available which thoroughly cover the subject. However, the experiencing of hydrodynamic action in bearings having no measurable geometric wedge or constrictions should not be thought of as an unusual occurrence. In the first place, truly flat surfaces are never found in practice. The irregularities themselves are minute converging and diverging wedges. The converging wedges tend to build up enormous hydrodynamic pressures. While the diverging wedges tend to build up negative pressures, the magnitudes are limited by the vapor pressure of the lubricant so that the net result is positive and the bearing surfaces can be separated by a film of oil. Also, Fogg (1, 2) explained the hydrodynamic phenomena in flat sliding bearings in terms of "thermal wedges". As oil is drawn between two surfaces in relative motion its temperature increases and thermal expansion takes place. This causes the pressure to increase and the bearing surfaces to tend to separate. Whatever the cause or causes experimental results have shown that hydrodynamic effects do take place in flat sliding bearings.

Molecular Adherence of Polar Lubricants

Polar additives are widely used because of their good boundary friction characteristics. These additives are characterized by an unsymmetrical molecule having a large electric moment. This causes the molecules to be strongly attracted to the metal surface where they either attach themselves or react chemically with the bearing material to form a surface film. The films thus formed have a low shear strength and, therefore, the bearing has low friction. In most cases this friction is independent of surface speed until hydrodynamic effect is initiated. Fatty acids have long chain molecules with carboxyl groups at the active end causing them to be strongly polar. Oleic acid is a typical fatty acid often used for experimental work.

ORIGIN OF THE PROBLEM

The most important characteristics of a machine tool slideway are long life and smooth and steady motion when operating over a wide range of speeds. Power loss due to friction is not of prime importance, as it is small in respect to the total power input. In fact, friction is often beneficial due to the damping effect.

When the coefficient of friction of a slideway decreases as speed increases, it is possible that unsteady motion will result. This condition depends upon the rate of change of friction with respect to speed, the mass of sliding member, the damping forces, and the rigidity of the drive train. By use of the correct lubricant additive, boundary friction can be made to be in dependent of sliding speed or even to increase as speed increases. However, the coefficient of mixed friction generally decreases as speed increases regardless of the additive used. Therefore, unsteady motion may occur in the presence of mixed friction conditions. If a decrease in friction with increasing velocity can be tolerated mixed friction conditions will greatly reduce the magnitude of friction and of wear. It is desirable to learn how various factors affect the transition from boundary to mixed friction so that these can be controlled to give the best results possible.

The object of this paper is to present data, somewhat exploratory in nature, concerning the effects of polar additive concentration, lubricant viscosity, surface finish, and load on the surface speed at which the transition from boundary to mixed friction takes place in flat sliding bearings. These data have been obtained by establishing a standard set of test conditions and varying one of the factors at a time. By observations made from these data general conclusion are drawn and presented.

APPARATUS AND PROCEDURE

Variable Speed Friction Apparatus

The variable-speed friction apparatus (Fig. 2) used in obtaining the experimental data for this paper, consists of five main components, each of which will be discussed separately.

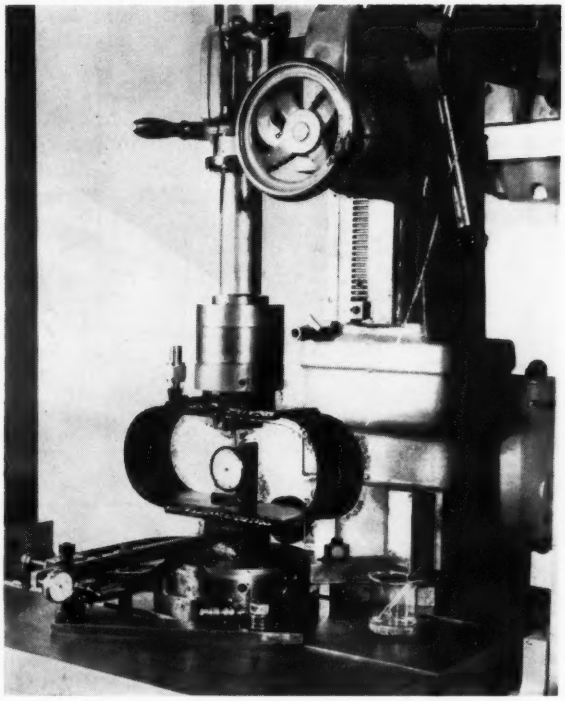

Fig. 2. Friction apparatus used in experimental studies.

1. Specimens

The specimens (Fig. 3) were designed to simulate machine tool slideways as nearly as possible. The rotating specimen is made of SAE 52100 steel, hardened to 63 Rc. This bearing surface is a plane. The rotating specimen is fixed to the spindle of a 21 inch sliding-head drill press. The stationary specimen is made of mechanite cast iron. This bearing surface is a land .250 inch wide and 1 foot in mean circumference. Six radial oil grooves were ground across the land. The bearings were ground and lapped.

2. Loop Spring

The apparatus has a calibrated loop spring dynamometer and a 0.0001 inch indicator for measuring the normal load applied by the manual feed hand wheel of the drill press.

3. Torque Indicator

A calibrated cantilever spring and 0.001 inch indicator are used to measure the torque produced by friction.

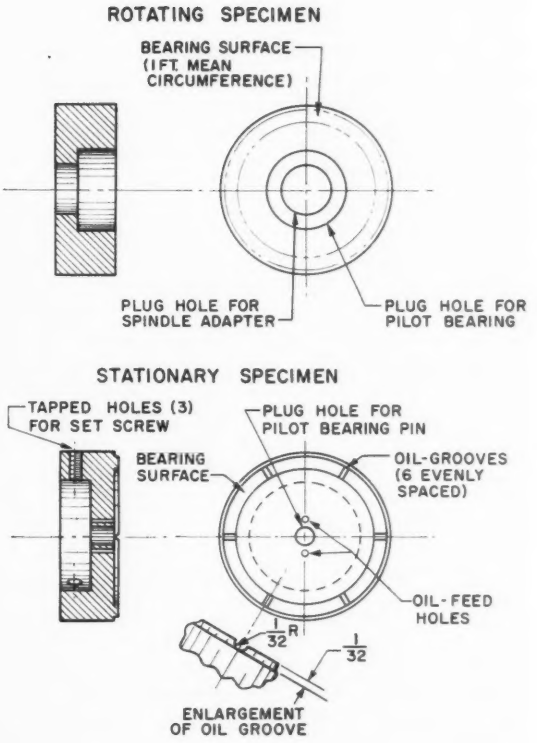

Fig. 3. Mating friction specimens used on variable speed friction apparatus.

4. Damper

A rotary damper was attached to the device in order to prevent oscillation at low speeds.

5. Motor Drive

The drive motor and control unit is a variable-speed type, having an effective speed range of 100 to 3200 rpm. The over-all maximum reduction ratio of the drive train was increased to about 4,000 to 1 by means of a jack-shaft, step pulleys, and gear reducer. The speed of the drill press spindle is infinitely variable from 0.03 to more than 2,000 rpm. Since the specimens have a mean bearing circumference of 1 ft., the sliding velocity in fpm is numerically equal to the spindle speed in rpm.

Lubricants

White paraffin oils of various viscosities were used as the base oil. A polar additive, oleic acid of USP grade, was used. This additive was chosen because it is a chemical compound widely used in experimental work.

Test Procedure

In obtaining the experimental data, a fixed procedure was used. The specimens were run at least eight hours after finishing to minimize the changing of the surface during the tests. For each test the surfaces

were scrubbed in degreasing solvent and then with acetone, thus removing the bulk oil and all but the more tenacious of the adsorbed films. The specimens were then run together dry under a load of 300 pounds at 20 fpm, to abrade the remaining adsorbed films, until the static friction coefficient rose to at least 0.3. This usually required only a few revolutions of the specimen. When the proper static friction value had been obtained, the surfaces were separated and the lubricant to be tested was applied with a medicine dropper. Great care was taken to prevent contamination of the lubricant and bearings. The specimens were then run together for one hour under load at a speed of 20 fpm, to give the active agents in the lubricant time to react thoroughly with the surface and to stabilize. Oil was added as necessary. The specimens were then stopped and quickly inspected to insure that no surface damage had occurred. The coefficients of friction were recorded at intervals of increasing speeds and then at intervals of decreasing speeds. Each test was repeated three times. The averages of the values obtained are presented here as results.

DISCUSSION OF RESULTS

All experimental results are presented in the form of graphs (Figs. 4 to 12). The results of all tests, except the varying load tests, are presented in the form of curves relating coefficient of friction and surface speed. The presence of oleic acid in the lubricant causes the boundary friction to be independent of surface speed. Since mixed friction is characterized by a falling curve the transition from boundary to mixed friction takes place at the speed at which the friction begins to decrease. When paraffin oil is used a curve is produced in which no boundary flat can be detected. One might at first conclude that there is no transition or that the transition takes place at too low a speed to be detected. However, evidence will be pointed out which indicates that there is boundary lubrication in this case and that a transition from boundary to mixed friction occurs at the same surface speeds as when oleic acid is present.

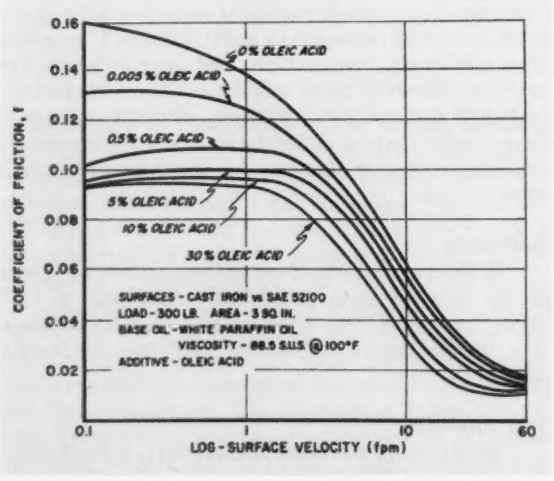

Fig. 4. Curves relating the coefficient of friction and surface velocity for various mixtures of oleic acid and paraffin oil.

Effects of Polar Additive Concentration

The results presented in Fig. 4 show that very small amounts of oleic acid in the oil greatly reduce the coefficient of boundary friction. Greater amounts of the additive have relatively little effect. This is quite similar to the data presented by Merchant (3), who attributed the above behaviour to the polar nature of the oleic acid molecule, which causes it to be preferentially adsorbed on metallic surfaces almost to complete exclusion of the essentially non-polar paraffin oil. Thus, the relative concentration of oleic acid at the bearing surfaces is very much higher than its concentration in bulk solution.

Fig. 5 shows that the speed at which the transition takes place is not significantly affected by the oleic acid concentration. The difference is slight enough to be explained by experimental error, changes in surface finish, and changes in viscosity due to the addition of the additive.

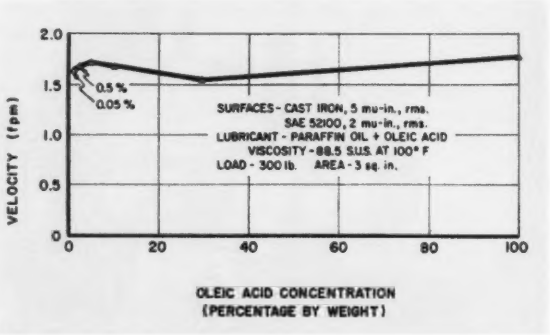

Fig. 5. Effect of oleic acid concentration on the surface velocity at which the transition from boundary to mixed friction occurs.

Effect of Viscosity

Fig. 6 shows that the static coefficient of friction and the coefficient of boundary friction are affected only slightly by lubricant viscosity. Fig. 7 shows that the surface speed at which the transition from boundary to mixed friction takes place is greatly reduced with increasingly viscosity. This is as would be expected considering Fuller's (4) explanation of mixed friction. Since mixed friction is a condition under which the load is partly supported by a fluid film and partially by direct contact, it is reasonable to conclude that a partial fluid film can be initiated with a higher viscosity lubricant at a much lower surface speed.

Effect of Surface Finish

The surfaces for this set of tests were prepared by lapping the surfaces together with various grades of lapping compound. The results shown in Fig. 8 indicate that the static and the boundary coefficients of friction are only slightly affected by surface finish. The speed at which the transition takes place is pointed out by the vertical dotted lines in Fig. 8. This speed is lower when the root mean square of the surface roughness is lower. This is as would be expected because the smoother surfaces tend to build a partial fluid film at a lower surface speed.

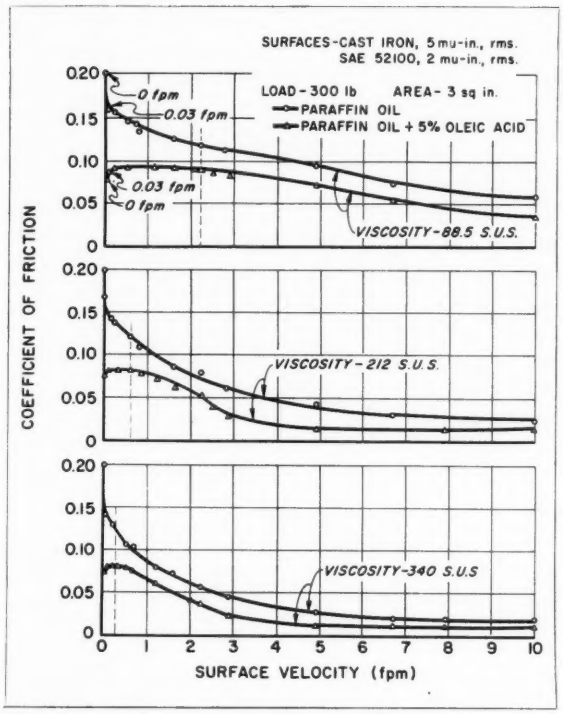

Fig. 6. Curves for oils of various viscosities relating the coefficient of friction to surface velocity.

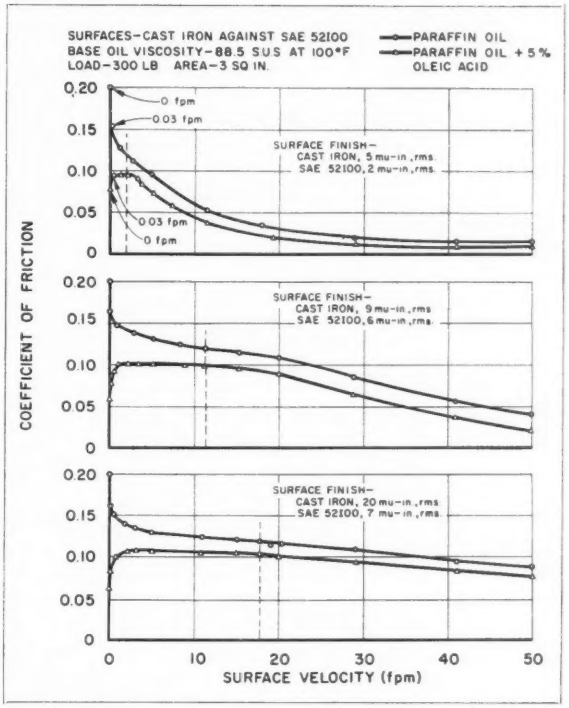
SURFACES-CAST IRON AGAINST SAE 52100
BASE OIL VISCOSITY-88.5 SUS AT 100°F
LOAD-300 LB AREA-3 SQ IN.
PARAFFIN OIL
PARAFFIN OIL + 5% OLEIC ACID
SURFACE FINISH-
CAST IRON, 5 mu-in., rms.
SAE 52100, 2 mu-in., rms.
SURFACE FINISH-
CAST IRON, 9 mu-in., rms.
SAE 52100, 6 mu-in., rms.
SURFACE FINISH-
CAST IRON, 20 mu-in., rms.
SAE 52100, 7 mu-in., rms.
COEFFICIENT OF FRICTION
SURFACE VELOCITY (fpm)

Fig. 8. Friction versus surface velocity curves for bearing of various lapped finishes. The velocities at which the transition from boundary to mixed friction occurs are indicated by dotted lines.

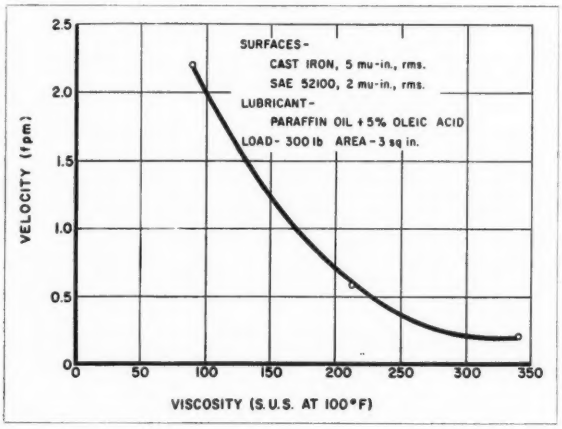

Fig. 7. Effect of varying oil viscosity on the velocity at which the transition from boundary to mixed friction occurs.

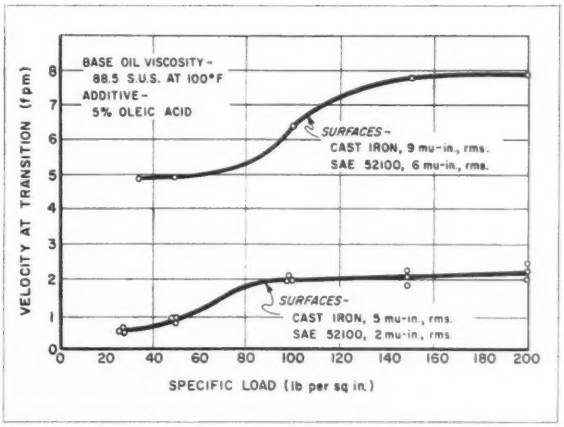

Fig. 9. Curves showing the effect of bearing load on the surface velocity at which the transition from boundary to mixed friction occurs. A curve is shown for each of two sets of bearings having different surface finishes.

Effect of Load

In this series of tests, data were not taken in the mixed friction region because the apparatus is not sufficiently accurate under light loads to measure low coefficients of friction. It should be noted that the possible error at the lightest load, 25 psi, is approximately 8%. The results are shown in Fig. 9. In general, the speed at which the transition occurs increases with load. When the load is higher, more lifting force is required to initiate a partial fluid film; therefore, it is reasonable to expect a greater speed to be required to obtain the transition.

Observation on a Commercial Lubricant

The lubricant under consideration is one commonly used on machine tool slideways. It contains a polar additive and a tacky additive for adhesive properties. It is effective in preventing stick-slip and metal-to-metal seizure under normal operating conditions. The results of the test of this lubricant, as shown in Fig. 10, are quite similar to those obtained when oleic acid is used in a base oil of approximately the same viscosity as that of the commercial lubricant. Although this oil has some extreme pressure properties and gives somewhat higher boundary friction than a lubricant containing oleic acid, the latter proves to be quite adequate for experimental friction work of practical nature.

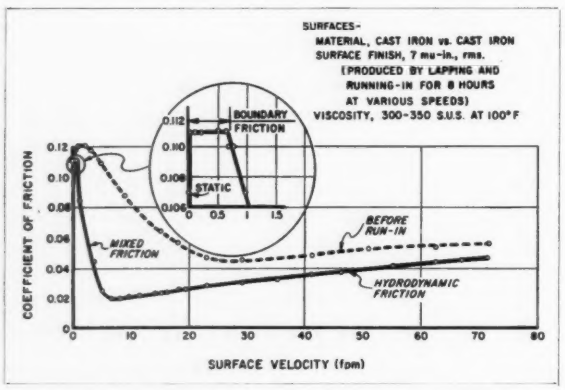

Fig. 10. Coefficient of friction versus surface velocity curve for a commercial lubricant.

ANALYTICAL STUDY

Since the bearings used for this work have no measurable geometric converging wedge, step or other constriction and because the films developed are probably too thin to exhibit bulk properties, the classical theories of hydrodynamic lubrication cannot be applied.

Many investigators have observed the conditions under which the load on a journal bearing was supported partly by fluid films and partly by direct contact areas of boundary lubrication. According to Fuller (4), G. B. Karelitz recognized these conditions as early as 1926 and suggested that the friction of total load carried by fluid films, during mixed friction conditions, could be estimated by considering the magnitude of the coefficient of friction. Although his work was concerned with journal bearings there is no reason why a similar method cannot be applied to thrust bearings.

Let P_H represent that part of the load carried by the oil films and P_F the part supported by direct contact areas. The total load, $P = P_H + P_F$. Then if f is the over-all coefficient of friction, f_H the coefficient of hydrodynamic friction and f_F the coefficient of boundary friction:

$$f\,P = f_H\,P_H + f_F\,P_F$$

$$f = f_H\frac{P_H}{P} + f_F\frac{(P - P_H)}{P}$$

$$\frac{P_H}{P} = \frac{f_F - f}{f_F - f_H} \qquad [1]$$

where $\frac{P_H}{P}$ is the fraction of the total load carried by the hydrodynamic films.

In order to make approximations of $\frac{P_H}{P}$ it is necessary to find means of determining f_F and f_H. The coefficient of boundary friction, when oleic acid is used, is almost independent of load and velocity. Therefore, f_F remains constant with respect to velocity. Fig. 11 shows a check of this. The data were obtained from specimens, after they had first been run with paraffin oil plus 5% oleic acid and then wiped dry. The coefficient, after the rise from the static, remains essentially constant with changing surface speed. It is reasonable to conclude that f_F, for a film caused by oleic acid can be taken as the value of the coefficient of boundary friction. The method of obtaining a value for f_F when no additive is used is somewhat more obscure. A review of the results shows that at the speed at which the transition to mixed friction occurs in bearings lubricated with paraffin oil and oleic acid, the value of the coefficient of pure paraffin oil is approximately 0.12. In

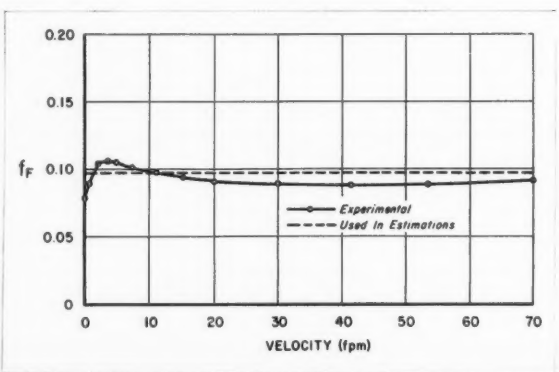

Fig. 11. Friction versus surface velocity curves for a film from oleic acid and paraffin oil.

some cases the coefficient of friction appears to be approaching a constant value before the transition occurs (see Fig. 8). Furthermore when a bearing which had previously been run with paraffin oil was wiped dry the coefficient dropped from the static value to a value of about 0.12 and remained constant with increasing speed. (See Fig. 12). Although not conclusive, the above is certainly evidence that under the same conditions of load, surface finish, and oil viscosity the transition from boundary to mixed friction will occur at the same surface speed when an additive is present as when an additive is not used. Also, the value of f_F can be considered to be 0.12 in this case.

f_H is approximated by assuming that at the point of minimum friction, f = f min., the bearings are completely separated by a fluid film and that the value is directly proportional to velocity. Then at a given velocity V:

$$\frac{f_H}{V} = \frac{f\ min.}{V\ min.}$$

Here V min. is the velocity at which f min. occurs. The above is only a rough approximation, but the accuracy of f_H is not critical at the lower speeds as it is small with respect to f.

From Fig. 8 f is 0.046 at 10 fpm when 5% oleic acid is used in the base oil having a viscosity of 88.5 SUS at 100°F and the surface finish of the cast iron specimen is 5 mu-in., rms. Here, f_F is 0.096. The minimum coefficient of friction is 0.012 occurring at 100 fpm. Then at 10 fpm:

$$f_H = \frac{10}{100} \times 0.012 = 0.0012$$

$$\frac{P_H}{P} = \frac{f_F - f}{f_F - f_H} = \frac{0.096 - 0.046}{0.096 - 0.001}$$

$$\frac{P_H}{P} = 0.526$$

Fig. 13 and 14 shows curves of P_H/P, calculated for various condition of viscosity and surface finish on bearings lubricated with oil containing oleic acid and oil not containing oleic acid. When

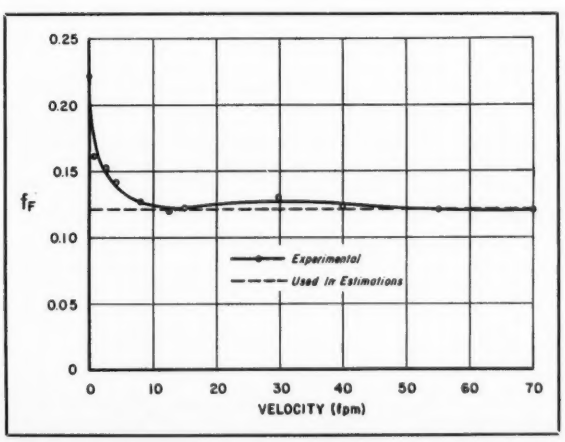

Fig. 12. Friction versus surface velocity curves for a film from paraffin oil.

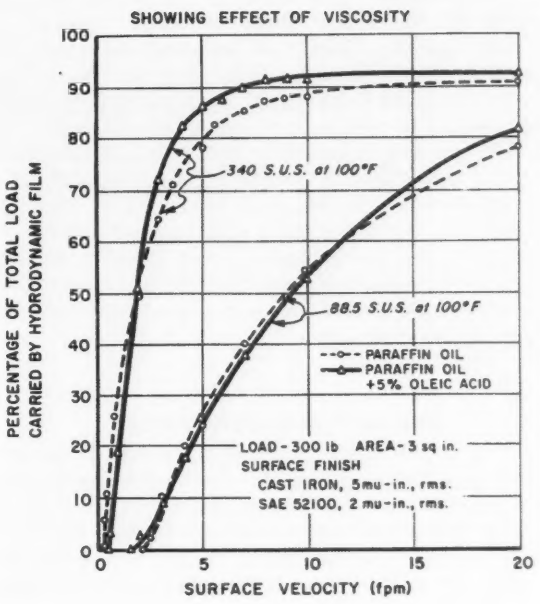

Fig. 13. Curves relating surface velocity to the load carried by hydrodynamic oil films in mixed friction conditions. The effect of viscosity of oil containing oleic acid and those not containing and additive is shown.

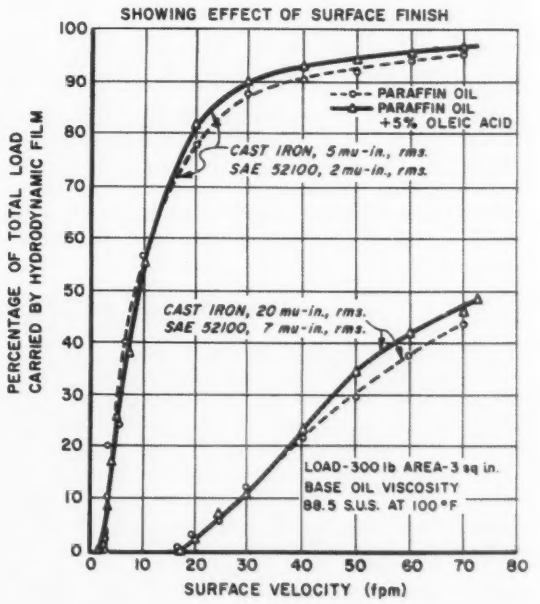

Fig. 14. Curves relating surface velocity with the load carried by hydrodynamic oil films showing effect of surface finish and the use of oleic acid as an additive.

the relative velocity is initiated the entire load is carried by the areas under boundary lubrication $P_H = 0$. As the velocity is increased the value of P is zero or essentially zero until the transition to mixed friction occurs. Then P_H/P increases rapidly with respect to velocity until almost all of the load is carried by a hydrodynamic oil film. It is most interesting that the curves for paraffin oil are essentially the same as those for paraffin oil + 5% oleic acid. This indicates that the use of oleic acid does not affect the amount of load carried by oil films. Figs. 13 and 14 illustrate the point that the factors tending to increase the value of P_H/P also tend to decrease the velocity at which the transition occurs.

SUMMARY AND CONCLUSIONS

The transition from boundary to mixed friction occurs at a lower surface speed with: (a) increasing base oil viscosity; (b) decreasing surface roughness; and (c) decreasing load. Also, it appears that the transition velocity is not affected by the addition of a polar additive to an oil, except where changes in the bulk viscosity or surface finish may result.

ACKNOWLEDGMENT

The author wishes to express appreciation to the entire research staff of The Cincinnati Milling Machine Co. for assistance in preparation of this paper. He wishes to thank specifically M. E. Merchant and C. O. Graves for helping to provide the technical and theoretical background and guidance in carrying out this research investigation.

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DISCUSSION

by Douglas Godfrey

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The decrease in coefficient of friction with increasing sliding velocity has been observed by other investigators under various conditions. At the writer's company we observed it on the Kinetic Oiliness Testing Machine (1); also, West found the effect on the low-speed friction apparatus (2). His results are shown in Fig. A1.

The reduction was observed at a velocity of 5×10^{-5} cm/sec which is about 0.0001 ft/minute compared with the lowest velocity of 0.03 ft/minute in the author's apparatus. Also, our constant value was 0.22 compared with the author's value of 0.12. Under the conditions of our experiments, which allowed considerable wear, we feel that the hydrodynamic component was very low. Perhaps in the author's appartus the decrease in friction with increased sliding velocity is not a result of a hydrodynamic film.

We have considered the explanations offered by Bowden and Tabor (3), Burwell and Strang (4), and Forrester (5) and decided that the effect is the result of complex surface phenomena and may only be explained by more refined experiments.

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Fig. A1. Friction of mineral oil (U.S.P.) in low speed friction apparatus.

AUTHOR'S CLOSURE

It is felt that Mr. Douglas Godfrey's remarks are in general accordance with the author's views. Referring back to Fig. 8, it will be noted that in the case of paraffin oil there are two areas in which the coefficient of friction decreases as the velocity increases. First in the boundary region, left of the dotted line, the effect may be explained by the complex surface phenomena referred to by Mr. Godfrey. Disregarding the difference in magnitude, Mr. West's results agree with the boundary conditions in Fig. 8. The higher magnitude of coefficients of friction can be explained by the difference in bearing configuration and material. The less severe conditions in the author's experiments did not allow considerable wear. It is only at the higher surface velocities that the decrease is attributed to the hydrodynamic components of mixed friction. The most convincing argument for this, besides those already given in the introduction, lies in the comparison of the familiar $\frac{ZN}{P}$ curves for journal bearings to the author's friction versus surface velocity curves. The existence of partial film lubrication can hardly be considered controversial in the former. What other mechanism could account for such similar results for the effects of viscosity and the reduction of the coefficient of friction to values below 0.02?

The author expresses thanks to Mr. Godfrey for his interest.

Load Support and Anti-Scuffing Properties of Lubricants IAE Gear Machine Studies

by R. J. Benzing

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Approximately two hundred and fifty (250) tests performed on the IAE 3¼ in. Center's Gear Machine using esters, mineral oils, and silicones are reviewed. Discussion of speed, temperature, and flow rate effects as well as repeatability and reproducibility is made. A rough correlation with the Ryder Gear Tester is presented for mineral oils and esters to use in guidance on development programs.

INTRODUCTION

Developments in the area of lubricants for high speed gearing in aircraft turbine engines have long required a clear and reliable index of the load-carrying capacity of the lubricant and dictated the use of load support evaluation tests. Many of the tests common to the field proved lacking in one or more aspects. To meet this need, studies were initiated in the early 1940's for the development of two lubricant gear evaluation machines which are of interest. One of these programs resulted in the gear machine developed by E. A. Ryder and commonly referred to as the Ryder Gear Test. The other was a study carried out in England using the Institute of Automobile Engineers Gear Tester first developed in 1935. The Ryder Gear Test found wide acceptance within the United States particularly for turbine oil evaluations, while the IAE enjoyed equal popularity in England and among other NATO nations. Both tests were and are used in specifications for turbojet engine oils as measures of load carrying capacity. The Materials Central, WADD, obtained an IAE Gear Tester in late 1955 from England for the purpose of evaluating its performance and to develop a possible correlation with the Ryder Gear Test.

The IAE machine is a hunting tooth type design with a 15 tooth driving gear and a 16 tooth driven gear. The test gear housing and test gears are shown in Fig. 1. The four square principle is used in the tester and the power return gears are double helical gears of 2¼ inch face width and 32 and 30 teeth. The normal speeds for the tester are 1000, 2000, 3000, 4000, and 6000 rpm obtained by various pulley arrangements on a 10 HP drive motor. The test oil and main oil supply are in separate systems so that the power return gears, bearings, and all parts of the power shaft drive system are not subjected to unknown oils. Table 1 is a tabulation of gear specifications for this machine.

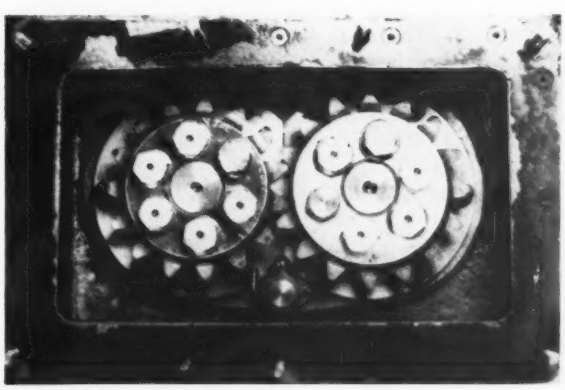

Fig. 1. Gear Housing.

EQUIPMENT, PROCEDURE, AND RESULTS

Before initial testing with the IAE rig was started, several shakedown runs were conducted to obtain experience in handling the machine. These runs indicated the need for several slight modifications of the test machine. These are tabulated below.

A. The glass in the observation window of the test gear cell was replaced with a high temperature resistant glass.

B. The test oil pump (1) and the test oil filter (2) were replaced by standard aircraft components to enable ready replacement with standard parts available in the United States.

C. Shaft seals in the test gear housing were replaced to prevent a slight tendency for leakage from the main oil supply to the test oil supply. The design for the new seals solved the problem quite satisfactorily although a slight leakage in the reverse direction is now experienced. With these modifications the test machine was felt to be suitable for extended testing. Exerience with the machine for the last three years has proved this to be correct. Little trouble has been experienced in its operation.

Approximately 250 IAE Gear Tests have been run on various oils to study such variables as speed, oil flow rate, and temperature. Both synthetic and petroleum oils were included. The tests employed a step-load type of procedure. That is, increasing loads were used until scuffing took place. In all cases an initial load of 10 pounds was applied through the torsion shaft and the test run for five minutes. The tester was stopped, the gears inspected, and if no scuffing was noted an additional 5 pound load was applied. The down time between each five minute run was also a five minute period. Increasing five pound loads were used in this manner until scuffing was noted. In many of the tests, scuffing is reported for both the approach and recess of the gear teeth. In these tests the load differs, and values for each are obtained in the same run.

TABLE 1
IAE Gear Specification

1. 15 tooth driving gear
2. 16 tooth driven gear
3. 3¼ inch centers
4. Nominal 5 diametrical pitch
5. Face width of 3/16 inch from tip to pitch line
6. Materials—2% nickel molybdenum steel to British Specification No. EN 34
7. Profile hardness not lower than 670 VPN (58 Rockwell C)

Scuffing of the gear may be observed in several ways.

A. Increase in motor load as shown on the ammeter.

B. Observation of "smoke" in the test gear housing.

C. Direct inspection of the gear teeth.

The first two cases are quite readily observed in tests where drastic scuffing is obtained. In those tests where the scuffing is not rapid and drastic, examination of the gears is required. Here care must be taken to insure that polishing is not being reported as opposed to actual tooth scuffing.

Each pound of applied load amounts to 61.043 pounds per inch of tooth face.

Table 2 presents the data for various oils as described in Table 3 by their chemical, Mean Hertz Load, viscosity and Four-Ball Wear characteristics. The operational characteristics for each test are given. The values given for scuff load are in pounds of applied load and should be multiplied by the factor 61.043 to convert to pounds/inch of tooth face. Where two values are given separated by a / (50/60 for example) the first is for the approach (app.) and the second for recess (rec.)

DISCUSSION OF RESULTS

Various test conditions as well as lubricant types were studied in this program. Effects of oil flow rate and temperature were studied using mainly MLO 57-101 (an uninhibited di 2 [ethylhexyl] sebacate). Oil types investigated included various synthetic and natural lubricants both neat and with additives. Speed effects were noted for most of the oils. Each of these will be discussed separately below along with data on reproducibility, repeatability, and correlation with the Ryder Gear Tester.

EFFECT OF SPEED

For the data presented in Table 2 a sharp decrease in load carrying capacity is experienced in the range of 2000 to 4000 rpm. Fig. 2 is a plot of load versus speed curves for several oils at 75 C and 270 ml/min oil flow rates. In the test, the load carrying capacity levels off after 4000 rpm. In the case of MLO

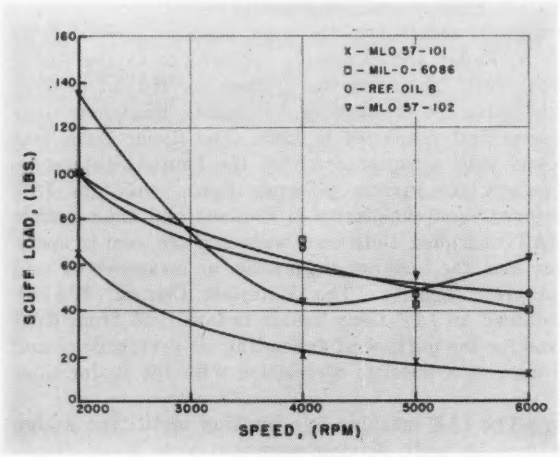

Fig. 2. Scuff Load Versus Speed For Several Oils.

TABLE TWO. IAE Gear Tests

RUN	SAMPLE	SPEED (RPM)	TEMPERATURE (°C)	OIL FLOW RATE (ML/MIN)	SCUFF LOADS†					
					RUN NO. ‡1	2	3	4	5	6
1	MLO 57-101	2000	75	270	65/65	65/65				
2			90	270	45/45	55/50	50/45	55/50		
3			90	370	50/50	50/50	50/50	60/60		
4		4000	75	270	15/15	20/20	25/25	20/20		
5			90	270	20/20	20/25	20/20	25/25		
6			150	270	15/15	15/10	15/10	15/15		
7			75	370	25/20	20/20	25/20	25/25		
8			90	370	20/25	30/30	25/25	25/25		
9			150	370	15/15	15/15	20/15	15/15		
10		5000	75	270	10/10	20/25	15/20	25/25		
11			90	270	15/20	20/15	20/20	20/15		
12			90	370	20/15	15/15	20/20	25/20		
13		6000	75	270	10/10	10/10	10/10	15/10		
14			90	270	15/10	10/10	10/10	10/10		
15			150	270	15/10	10/10	15/15	10/10		
16			75	370	10/15	10/15	10/10	10/15		
17			90	370	15/15	15/20	10/15	10/10		
18			150	370	10/10	10/10	10/10	10/10		
19	MLO 57-102	2000	75	270	120/120	150/150				
20		4000	75	270	45/45	40/40	45/45			
21		5000	75	270	55/55	55/55				
22		6000	75	270	80/80	45/45				
23	GTO-133	2000	75	270	95	100				
24		4000	75	270	60	60				
25		5000	75	270	30	35	35	35	40	
26		6000	75	270	35	15	35	35	40	
27		1000	75	270	215					
28		5000	150	270	30	30	30	30		
29	GTO-38	5000	75	270	90	95	75	85		
30	GTO-135	1000	75	270	215					
31		2000	75	270	150	140	115	145		
32		4000	75	270	85	80				
33	GTO-135	5000	75	270	90	80	80	80	80	75
34		6000	75	270	65	70	60	65	70	
35		5000	150	270	85	75	95			
36	O-56-11	2000	75	270	80	80				
37		6000	75	270	30	45	40			
38	O-56-62	2000	75	270	90	90	90*			
39		4000	75	270	50	55	55*			
40		5000	75	270	40	45	45	35	45*	
41		6000	75	270	30	40	25	30	35	35*
42	90% MLO 58-88 +10% Silicone	6000	75	270	50**	55**				
43	Silicone	2000	75	270	40	35	60*	60*	60*	65*
44		4000	75	270	30	25	35*	30*		
45		5000	75	270	25	20	25	25	20*	25*
46		6000	75	270	15	15	20	10	15*	15*
47	RDE/O/305	2000	60	½ Imp. pt.	65	90	70	80		
48		4000	70	1 Imp. pt.	45	35	50	45		
49		6000	110	1 Imp. pt.	40	35	30	30		
50	VMTOEP/B848/ CLW 1211	2000	60	½ Imp. pt.	180	185	180	135***		
51		4000	70	1 Imp. pt.	85	90	80	90		
52		6000	110	1 Imp. pt.	60	60	55	65		
53	PPO 500	2000	75	270	105*	90	85			
54		4000	75	270	45	50	45	50		
55		5000	75	270	45	50	40	40	40	45
56		6000	75	270	35	30	30	35	30	
57	Ref. Oil B	2000	75	270	100	100				
58		4000	75	270	70	65	70*			
59		5000	75	270	50	55	55*	50	55	
60		6000	75	270	50	55	55	50*	50	
61	MIL-O-6086	2000	75	270	95	100	100	105	105	100
62		4000	75	270	55	60	60			
63		5000	75	270	45	40	45	40	85	45
64		6000	75	270	40	40	35	45	40	40

† App./Rec.

‡Individual runs—New gear face in each run—Two runs per set of gears—Same batch of oil.

*Check Run **May be Contaminated ***Oil Flow Questionable

TABLE 3.—IDENTIFICATION OF OILS

		MEAN HERTZ LOAD	VISCOSITY 100 F (CS)	VISCOSITY 210 F (CS)	4-BALL WEAR, 75 C 2 HRS.—1200 RPM (MM SCAR)	
					4 KG	40 KG
MLO 57-101	Di 2 ethylhexyl sebacate	19	12.8	3.3	.55	1.03
MLO 57-102	MLO 57-101 + 6% dioctylphosphite	39	—	—	.33	.54
GTO-133	Specification MIL-L-7808 oil (Diester Base Oil)	36	12.6	3.3	.25	.45
GTO-38	Polyglycol formulated to Specification MIL-L-7808	30.5	11.8	3.16	.29	.37
GTO-135	Same as GTO-38—second batch	30.5	—	—	.29	.37
0-56-11	Specification MIL-L-25336 oil	21	12.8	3.4	.56	1.07
0-56-62	Same as 0-56-11—second batch	21	—	—	.56	1.07
Silicone	Phenyl methyl silicone	12.5	53.8	18.6	1.07	3.65
MLO 58-88	Complex ester	—	—	—	—	—
RDE/O/305	Low load-carrying petroleum oil	—	—	—	—	—
VMTOEP/B848/ CLW 1211	High load-carrying petroleum oil	—	—	—	—	—
PPO-500	Specification MIL-O-6082 Grade 1065 oil, mineral oil	19	103	11.2	.31	.92
Ref. Oil B	Specification MIL-O-6082 Grade 1100 oil, mineral oil	—	238	20.3	.30	.31
MIL-O-6086	Petroleum based gear oil, Specification MIL-O-6086 Grade M	—	60-82			

57-102 an increase in load carrying capacity is noted for the 5000 and 6000 rpm value over that of 4000 rpm. This may be due mainly to the fewer number of runs as well as a larger spread in data. However, increasing load carrying capacity has been noted by other investigators (4, 5, 6, 7) although at much higher speeds, in region of 10,000 to 15,000 rpm. Borsoff's (5) data does show reversals in the load-speed curves in the neighborhood of 5000 rpm for several mineral oils and synthetics. No studies in this area on the IAE Tester are contemplated in as much as the 6000 rpm top speed of the tester is limiting for the purpose of extending such studies into the region where the effects are most commonly found.

For future evaluation a speed of 5000 rpm is recommended for screening studies as this is well into the region of the flat portion of the curve. Where the effect of speed is to be determined on a given oil, runs at 2000, 4000, 5000, and 6000 rpm are recommended.

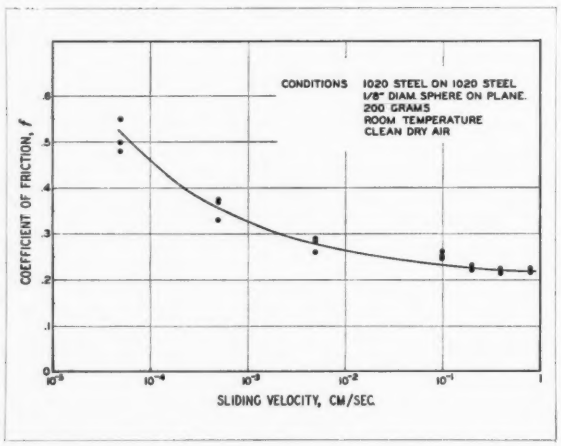

Fig. A1. Friction of Mineral Oil (U.S.P.) in low speed friction apparatus.

EFFECT OF LUBRICANT FLOW

In all tests the lubricant jet was directed at the meshing of the gear as shown in Fig. 1. Borsoff (5) as well as others has shown that this results in the highest load carrying capacity. The effect of flow rate was investigated using MLO 57-101. Table 4 presents the data for flow rates of 270 and 370 ml/min. at various conditions. In most cases a slight increase is noted in load support, however, this is not a major change and probably indicates that 270 ml/min. is a suitable flow rate for future work. At the flow rates of this investigation, minor variations in flow volume had no significant effect on the scuff load.

EFFECT OF TEMPERATURE AND VISCOSITY

Various investigations have reported the results of viscosity and/or temperature effects on gear load support in the Ryder, IAE, or similar equipment (1, 5, 6). Tables 4 and 5 show little effect of temperature on the load support values of the various lubricants at a fixed pitch line gear velocity. The data in Table 4 is for an oil consisting only of a base stock, whereas the oils used to obtain the data in Table 5 contained a blend of additives including anti-wear agents in a diester base stock. For the oils in Table 5 this may be a balancing of loss due to loss in viscosities (from about 5 cs to 2 cs over the 75 C to 150 C range)

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and an increase due to increased additive effects. The increase for MLO 57-101 at 90 C, although slight, appears to hold at both 4000 and 6000 rpm and can not be explained other than by increase in chemical reactivity of the oil opposing a viscosity effect. Table 6 demonstrates a decided increase in load support with increased viscosity for two similar oils (PPO 500 and Ref. Oil B). Greenwood and Morton (1) show decidedly different levels but the same general effects of load support versus viscosity for synthetics and mineral oils in the IAE machine. These investigators show quite drastic losses with increased temperature. The data on the two non-additive mineral oils (Table 4) and the data of Greenwood and Morton indicate that the supposition (that of counterbalancing effects of viscosity and chemical effects of additives) is correct for diester oil, MLO 57-101 and the oils listed in Table 5. To study temperature effects thoroughly, a more detailed program on non-additive oils is indicated.

TABLE 4.—Effect of Flow on Load Support with MLO 57-101

SPEED (RPM)	TEMPERATURE (°C)	SCUFF LOAD #*	
		270 ML/MIN	370 ML/MIN
2000	90	47.5	52.5
4000	75	20.	21.25
	90	21.25	25.00
	150	17.5	16.25
5000	90	16.25	17.50
6000	75	10.00	10.00
	90	10.00	12.50
	150	11.25	10.00

#Diester Base Oil.
*Using average of minimum values for approach on recess from Table 2.

TABLE 5.—Effect of Temperature on Load Support

OIL	SPEED (RPM)	SCUFF LOAD (#)	
		75 C	150 C
GTO-133	5000	35	30
GTO-135	5000	80	85

Diester Formulation Containing Additives.

TABLE 6.—Load Support Data for Two Mineral Oils ***

SPEED (RPM)	FAILURE LOAD FOR REF. OIL B* (POUND)	FAILURE LOAD FOR PPO 500 ** (POUND)
2000	100	93
4000	68	48
5000	53	43
6000	52	32

***Temperature = 75 C.
*93-103 cp at 210 F.
**62-68 cp at 210 F.

REPEATABILITY AND REPRODUCIBILITY

Repeatability of results within this Laboratory has been very good. The majority of the runs have been made on the same machine with the same operator so that a good analysis can be made. Calculated standard deviations divided by the average score load and multiplied by 100% have run between five and ten percent for the most part. This is in good agreement with results published (1) and an overall estimate of repeatability given by another user (8) of this type test. The British Institute of Petroleum in conjunction with various users of the IAE Gear Tester conducted a correlation program on the machine. Although published results are not available, the minutes of the correlation study have been obtained and a summary will be made of the program. In all there were fourteen participants including this Division. The test employed was generally the same as previously described except that specific cleaning methods, lubricants, etc., were to be used. Three speeds (2000, 4000, and 6000 rpm) three temperatures (60, 70, and 110 C) and two oil flow rates (½ and 1 imp. pt/min) were employed using oils RDE/0/305 and VMTOEP/B848/CLW 1211 from Table 2. Each test was conducted in quadruplicate. The results of this, although not completely finalized, gave repeatability of 4.5% of mean load + 7 pounds and reproducibility of 14% of mean load + 22 pounds. This would tend to give a higher percentage spread at the lower loads than at the higher loads and this has been the experience in this Laboratory. However, these values would give poorer percentage repeatability at the lower loads than is normally experienced at this Laboratory.

CORRELATION WITH OTHER TESTERS

One of the purposes of this program has been to correlate results with the Ryder Gear Tester.

Table 7 presents results for the various oils evaluated both in the Ryder Gear Machine and IAE Gear Tester. The four oils marked with an asterisk show a reasonable degree of correlation at 5000 rpm. They included three petroleum oils and an additive diester.

MLO 57-101 and MLO 57-102 deserve special comment. MLO 57-101 is di 2 (ethylhexyl) sebacate and when run in the Ryder Gear Test gave a value of 2730 pounds/inch of tooth. However, other reported data on the same material give values of approximately 2200 pounds/inch of tooth width. Although the value of 1200 pounds/inch of tooth obtained in the IAE Tester does not approach this value it at least indicated low load support as shown in both testers. MLO 57-102 is an additive diester and shows the additive effect in the IAE. The test in the Ryder Gear Machine gave low results but there is some indication that the additive may have settled out and only the diester was tested in this run. The operator was not familiar with the material and did not agitate the can before taking his sample from the top.

The lack of correlation of the silicone and 0-56-62 can not be explained due to the few number of tests. However, such differences as sliding speeds and addi-

tive reaction on the various steels may well be the cause.

It is felt that use of a 5000 rpm test on the IAE tester would give fair correlation with the Ryder on esters and mineral oils as long as unusual additive response due to possible differences in gear materials are not encountered. Such effects could be noted in more simple bench tests. No direct correlation for specifications can be drawn, however, and the result of any development program carried out on one tester should be checked on the other if qualification is desired under the specification using the second rig. Each tester has given satisfactory screening results for the engines used by the particular country and quite likely either would serve well as long as the proper gear materials were used.

TABLE 7.—Correlation of Ryder Gear Machine and IAE Tester
Flow Rate = 270 ml/min
Temperature = 165 F (Ryder)
158 F (IAE)

Oil	Ryder Results 10,000 rpm (Failure Load #/inch of tooth)	IAE Results (Failure Load #/inch tooth)			
		2000	4000	5000	6000
MLO-57-101	2730 (2200)	3970	1220	1220	610
MLO-57-102	1975	8230	2620	3350	4150
Ref. Oil B	2690*	6100	4150	3250*	3170
Silicone	3730	3230	1830	1400	915
PPO 500	2300*	5670	2930	2620*	1950
GTO-133	2160	5980	3660	2135*	1950
O-5 6-62	4120	5490	3230	2560	2050
MIL-O-6086	3500*	6150	3540	3050*	2440

* (See Section VIII)

CONCLUSIONS

The IAE Gear Machine is a reasonably simple and reliable test for evaluating the load support characteristics of oils. It is felt that the repeatability and reproducibility for this type test is very good. A minimum of instruction is required to train the operator before he becomes proficient.

Although there is not a direct correlation with the Ryder Gear Machine it is felt that results at 5000 rpm in the IAE are sufficiently close to be indicative of Ryder results on esters and mineral oils. Differences may be caused by additive effects due to a difference in metallurgy or rotational and sliding speeds. Oils could be developed for a specification using one of the two tests with the other machine but should be checked periodically to determine possible unusual formulation packages.

Because of its simplicity of operation the IAE Gear Machine makes a good screening tool. There is actually less time in running tests on it than in a bench test of the magnitude of the Mean Hertz Load.

Speed and flow rate effects can be readily studied but due to a low temperature limitation on the unit in this Laboratory no sound program can be set up for temperature investigations except for below 300 F.

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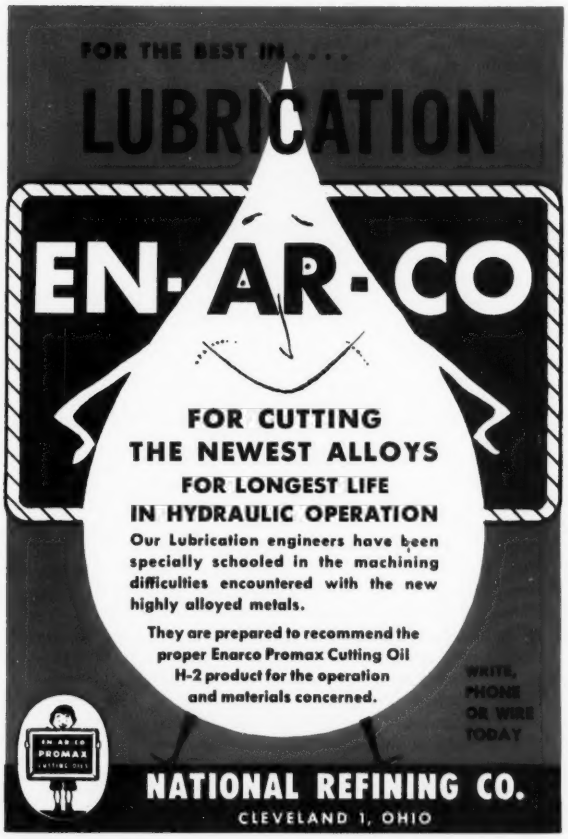
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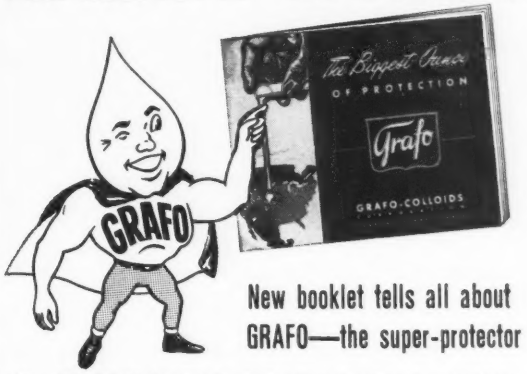

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Lubrication Abstracts

Edited by W. E. Campbell

Selected literature compiled by members of the Technical Committees and Industry Councils of the Society.

Ehrhard Mayer (Borg-Warner Research Center, Des Plaines, Ill.), "Leakage and Wear on Mechanical Seals", *Machine Design*, v. 32, no. 5, 1960, pp 106-13.

An investigation is described to determine the wear and leakage rate with unbalanced mechanical seals for 20,000 hours with five face material combinations and three different fluids. The object is to establish major causes of the difficulties, determined corrective actions, and, if possible, formulate methods of calculating leakage rates. Results indicated that no fluid pressure exists between seal faces, in contrast to previous theories. Leakage, wear and friction are determined by boundary lubrication. Surface width appears to have no influence on leakage. Leakage varies with the square of the distance between mating faces and inversely with the square of the face pressure. No influence of viscosity of the liquid is noticed, and leakage is the same for all fluids but nearly 100 times greater for external than internal rotating seals. *(Abstractor: V. E. Vorhees)*

by Vernice Hopkins, "Modification of Shell Four-Ball Wear Tester for 800°F. Operation and Development of a Lubricant Evaluation Technique," Rept. for Apr. 57-Dec. 58 on Aviation Lubricants, Contract AF 33(616)5205 June 59, 68 p. 4 refs. WADC Technical Rept. 59-48: AD-215 447.

Modifications made to adapt the Shell four-ball wear tester for 800° F. operations are outlined. Detailed discussions of the design changes, the frictional torque measuring setup, lubricant circulating system, and the application of additional controls and instrumentation are presented. The evaluation of the performance of the modified wear tester is described. Operating test procedures for a conventional test and the alterations required for lubricants solid at room temperature and for lubricants circulating through the ballpot assembly are included. Testing experiences are discussed, especially for high temperature tests. A method for evaluating lubricants is outlined. *(Author Abstract per Robert McKay, III)*

John H. Laub (Jet Propulsion Laboratory, California Institute of Technology, Pasadena, California), "Hydrostatic Gas Bearings", ASME—Paper 59-Lub-1 for Meeting October 20-22, 1959.

The pressure distribution in the case of radial outward flow of gas between two flat circular discs is derived under the assumptions of laminar, isothermal, compressible flow with no entrance or exit losses. Integration of this expression gives the load-carrying capacity for a step bearing utilizing such gas flow in terms of the inner and outer radii and the pressures at these radii. The pressure at the inner radius is then found by equating the flow through a feed orfice to the flow in the bearing gap in order to permit calculation of the load capacity in terms of the plenum (supply) pressure and the bearing gap height. Experimental data is given showing excellent agreement between calculated and observed values of recess pressure, flow, and load for various ratios of radii and for various gap heights.

It is concluded that the theoretical expressions derived for the bearing gap, flow and load capacity describes the performance of hydrostatic pad and step bearings very accurately within the range of pressures (8 to 48 psig.), step radius ratios (2 to 36), and for the particular orifice size used. The latter was such as to limit the flow to less than 0.003 cfm even at the largest gaps. *(Abstractor: V. H. McNeilly)*

H. S. White (National Bureau of Standards), "Friction and Endurance of Prelubricated and Unlubricated Ball Bearings at High Speeds and Extreme Temperatures," *Journal of Research*, v. 63C, no. 1, 1959, pp. 19-29.

Endurance tests are made with 20-millimeter bore prelubricated ball bearings at 10,000 revolutions per minute, using polyester felt rings saturated with di (2-ethylhexyl) sebacate containing appropriate additives, at ambient temperatures ranging from 80° to 400° F. Tests are made with greases for comparative purposes. Results indicate that endurance with these oil-soaked pads is comparable with the high-temperature performances of MIL-G-3278 grease at 250° F. and MIL-G-3545 grease at 300° F. Some endurance tests are made with special bearings made of high speed tool steel, prelubricated at 450° F. ambient temperature, and unlubricated at 650° F. ambient temperature. Experiments are made with greased bearings and with felt-pad oiled bearings in gyros at high temperatures. With oiled bearings and unlubricated bearings, wearing or breaking of separators (ball cages) is the most common cause of failure in these tests. *(Abstractor: R. A. Schoeppel)*

O. Emicke, "Roll Pressure and Life of Anti-Friction Bearings for Hot and Cold Rolling Mills," (IN GERMAN) *Neue Huette*, v. 4, no. 3, 1959, pp. 174-78.

The fundamental relationship between load rating (bearing capacity), applied load and speed, and calculated minimum life is given. A nomogram simplifies the calculations and includes other dependent variables such as: shaft diameter and length, resultant bending stress, rolling speed and roll diameter. Examples show the use of the nomogram. *(Abstractor: M. Blume)*

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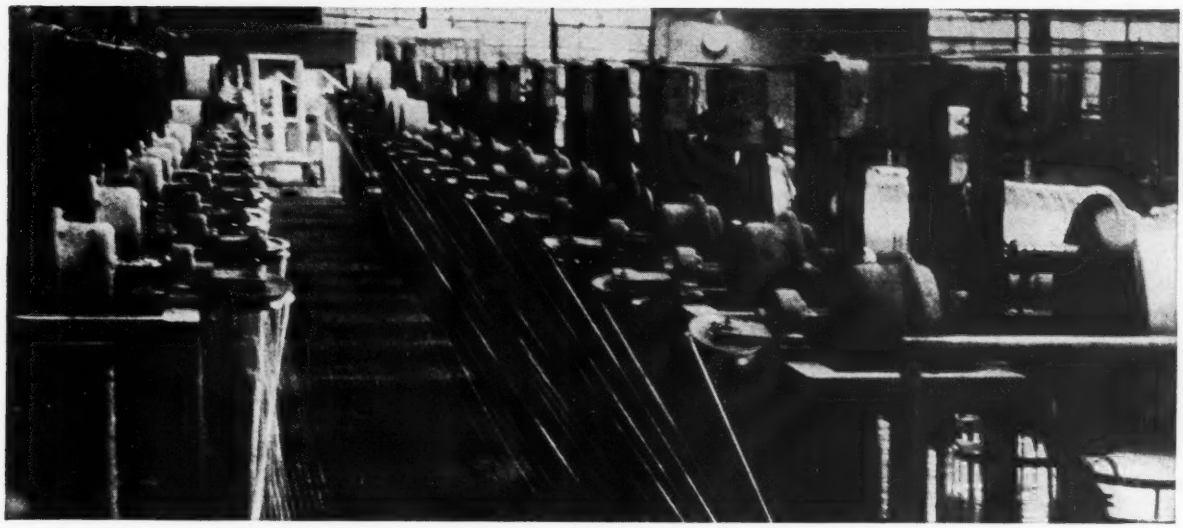

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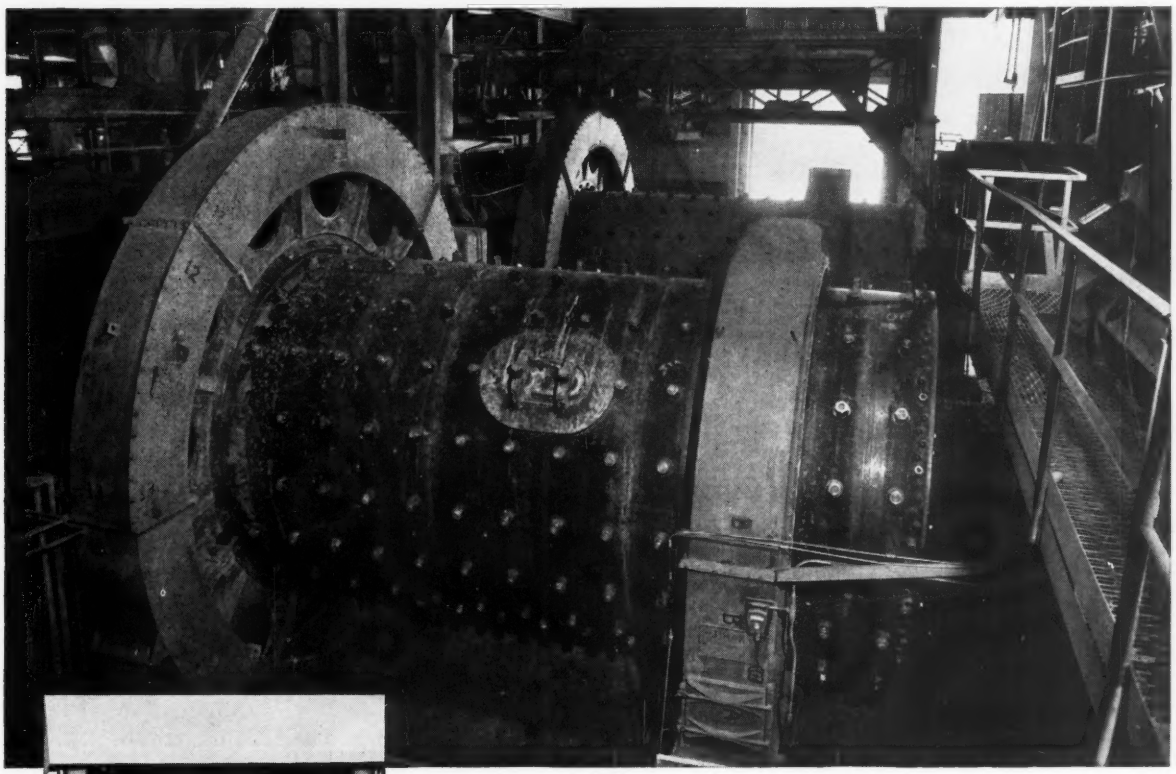

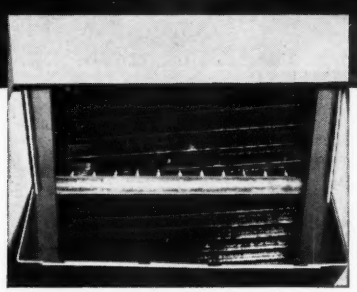

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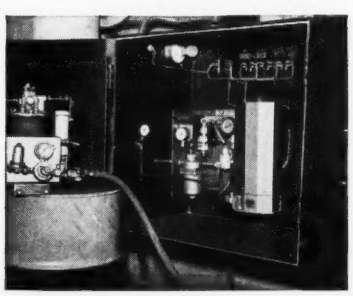

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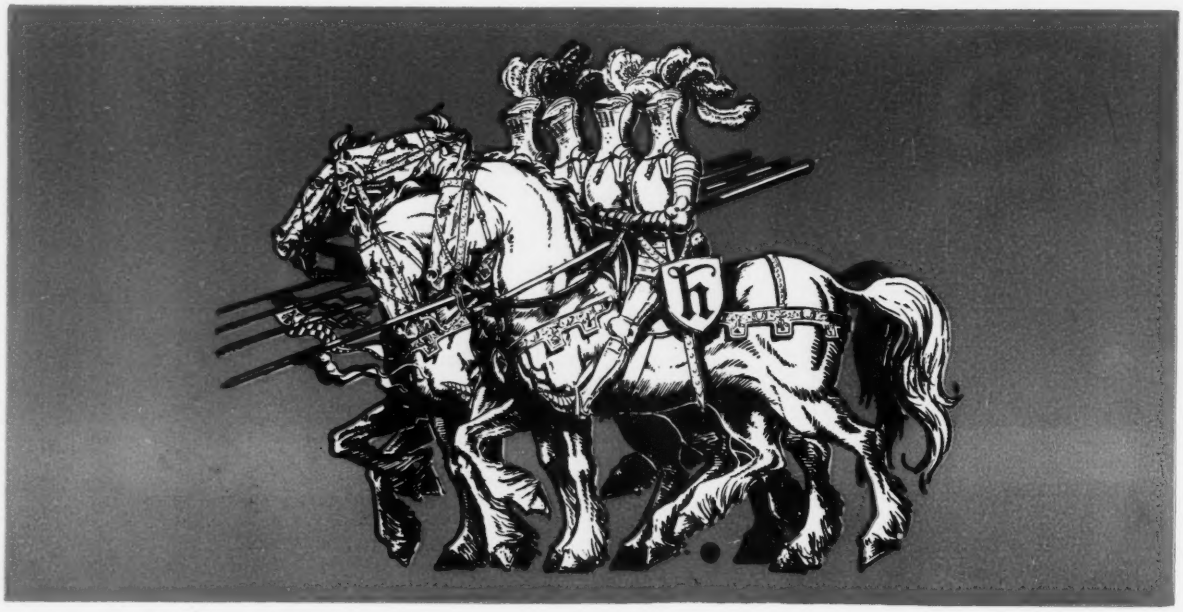

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